HANDBOOK
OF
MECHANICAL REFRIGERATION

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BY

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PREFACE

In the author's opinion an engineering handbook must naturally include all information an engineer is likely to be in need of in the particular field under discussion. It should have some theory as the basis for the fundamental formulæ, and it certainly must have all information required for the comprehension of the design and the practical operation of any system falling within the field of the handbook. In the present case, therefore, there is given a brief treatment of certain thermodynamic material in the expectation that for more detailed study of this phase of the subject the reader will consult a text on thermodynamics; also, such information concerning electric motors, steam and oil engines, and related matters as is absolutely essential for the grasp of the main material.

Much of the material of this book has been gathered and used for teaching purposes. This is particularly true of the special applications of refrigeration, a division of the subject which, therefore, has been treated perhaps more fully in this book than has heretofore been attempted. The material on compressors and on the absorption machine has been brought up to date, and it is thought that the chapters on cold storage and ice manufacture will be found particularly complete and of practical value. A recent great increase in the use of the household refrigerating machine and the so-called ice cream cabinet has materially assisted in the development of the automatic machine, while the more or less general fear of ammonia has been the incentive towards the use of some safer refrigerant such as methyl chloride and the hydrocarbons. We have more and more such things as "manufactured weather," the use of precooling and refrigeration during transit, district cooling, hotel and apartment house refrigeration. All of these subjects will be found treated in the chapter on cold storage. The development of the raw water ice manufacture has stimulated the study of water treatment and air agitation while the increasing cost of power and labor has necessitated the use of the Diesel oil engine, the electric motor and labor-saving devices in the tank room. Material, therefore, on these matters also will be found in this volume.

Wherever possible, illustrative examples have been worked out. As a rule such illustrative problems are very similar one to another, and vary only in the physical constants used. In some special cases these constants are not known accurately (as, for example, the heat capacities of distillates which vary in the different oil fields). Not many years ago Professor Denton was forced to find out experimentally for himself certain properties of ammonia before he could work up his tests; and the early refrigerating engineers experimented on the specific heat of various commodities, such as plowshares, brines, etc., before calculating the refrigeration required in special problems. In some cases similar procedure is still necessary, especially in certain chemical processes, but through the efforts of the American Society of Refrigerating Engineers, the United States Bureau of Standards, and other societies, refrigerating data are being rapidly collected and distributed. Where such information is already available the author has endeavored to embody it in this treatise.

An attempt has been made to give the authority for all material not the author's own, and to include references whenever possible. In particular it should be stated that the introduction follows closely certain parts of the "Principles of Thermodynamics," by Professor G. A Goodenough, to whom thanks are also due for many helpful suggestions. The manufacturers of compressors and refrigerating equipment have been very generous in supplying information without which a book of this character would not be possible. To these engineers, and in particular to his former students, now engaged in the refrigerating industry, the author desires to express his thanks. He wishes also to thank Mr. R. E. Gould, research assistant in the Engineering Experiment Station at the University of Illinois, for his reading of the page proof.

With a book of this nature, especially in the first edition, it is practically impossible to eliminate errors. The author solicits information regarding any defects in the problems or the text.

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HANDBOOK OF REFRIGERATION

CHAPTER I

INTRODUCTION

Mechanical Refrigeration is one of the many applications of the theory of heat energy, but it is a reversed cycle as compared with that used in prime movers. In mechanical refrigeration heat is absorbed at the lower temperature and is rejected at the upper temperature, and a pump—the compressor or other device of similar nature—is employed to make the cycle possible. A book on refrigeration has to cover primarily the field of its application, but it needs also to give material allied with the industry and which the refrigerating engineer finds need of in his work as well as the fundamental basis for the cycles and the formulæ used. In this chapter, therefore, these related matters will be taken up very briefly, and an explanation of the diagrams used in representing the refrigerating cycle will be given.

Energy.—A moving body, a compressed spring, or a substance like steam is said to possess energy when it has the ability to perform work while its condition undergoes a change. This energy may be mechanical *kinetic* energy due to its motion, or mechanical *potential* energy due to its position or its state of aggregation. When the energy of a substance is due to the motion or the change or arrangement of the molecules this energy is called *intrinsic* energy.

Temperature.—Temperature is an indication of the rapidity of the vibration of the molecules of a body, and usually a change in the intensity of this vibration is evidenced by a change in the volume of the body, as when heated air expands or cooled mercury contracts. There are two scales of temperature in common use, one in which the distance between the melting and boiling points of water is designated as a hundred divisions (with the zero and the one hundred at these two points, respectively), and the other naming 32 as the freezing and 212 as the boiling temperature with 180 divisions between. In the United States the

latter, the Fahrenheit scale, is used with practical unanimity by mechanical engineers; while the other, the Centigrade scale, is used in Great Britain and on the Continent. The *absolute zero* of temperature, as is evident from what precedes, is the condition where there is no vibration of the molecules of the body. This point is considered to be 459.7 degrees below the zero on the Fahrenheit scale.

Pressures.—Pressures may be measured in inches of water, or of mercury, in atmospheres, or in kilograms per square centimeter. The custom in the United States is to use as a unit the pound per square inch. The ordinary gage records in pressures above the atmosphere, so that the *absolute pressure* is obtained by adding to the gage reading the equivalent of the atmospheric pressure.

Heat.—It has been found by experiment that if two bodies of different temperature are placed near one another heat will flow from the hotter to the colder body until finally the same temperature is attained by each. The heat in a body may be *sensible* heat (indicated by its hotness or lack of hotness) or it may be *latent* heat, a term used to denote the change of aggregation of the molecules such as is evident when the substance changes from a solid to a liquid, from a liquid to a gas, or vice versa. The unit of heat is taken as the British thermal unit (B.t.u.) which is defined as $\frac{1}{180}$ of the amount of heat required to raise one pound of water from 32 to 212 deg. F.

There are other forms of energy besides heat energy: such as, chemical and electrical energy. By numerous experiments it has been demonstrated that energy of one kind can be transformed into energy of any other kind.

The Energy Equation.—Experiment has proved that if heat energy is supplied to a substance a change may occur in the intrinsic energy, or external work (work performed on an external body) may be done, or both. This is stated as follows:

$$_1Q_2 = U_2 - U_1 + AW$$

where $_1Q_2$ is the heat added during the process *12*, U_1 is the initial and U_2 is the final intrinsic energy of the substance and W is the work performed in foot-pounds (ft.-lb.). The expression A is for the reciprocal of the mechanical equivalent of heat ($\frac{1}{778}$).

The intrinsic energy may consist of both thermal *kinetic* energy, indicated by the temperature, and thermal *potential* energy, indicated by the state of aggregation of the substance. Energy, like pressure, volume, temperature, etc., being a point function, may be represented by a point on a coordinate plane and is independent of the path by which

the stated condition was obtained. It follows from this that in a complete cycle, on arriving at the initial point, the *energy change* is zero.

The First Law of Thermodynamics.—The experiments of Rumford and others have proved that mechanical and heat energy have a certain relationship; that is, when work is expended in producing heat, the heat produced is proportional to the work performed. This can be expressed by the relation:

$$Q = AW$$

where W denotes the work performed, usually expressed in mechanical engineering in foot-pounds (ft.-lb.). The quantity AW would then be in B.t.u.

Conversion Constants.—The following conversion constants, based on the first law of thermodynamics, will be of value in the solution of problems.

1.0 hp.	= 42.44 B.t.u. per minute.
	= 2546 B.t.u. per hour.
1.0 KW. hr.	= 3414.5 B.t.u.
1.0 B.t.u.	= 777.6 ft.-lb.

Specific Heat.—The ability of different substances to absorb heat varies, as instanced by water, copper and iron. The ratio of the capacity for the absorption of heat possessed by equal weights of certain substances, as compared with the capacity of *water*, gives the specific heat (c), which is seldom more than unity. In the case of gases and vapors it is necessary to state under what conditions the heat was absorbed, as for example whether it was at constant pressure (c_p) or under constant volume (c_v).

The *p-v* Diagram.—Changes in a substance involving changes in pressure and volume may be represented by a line on the plane of two coordinate axes, as for example the so-called *p-v* plane. It can be shown that the projected area under such a " curve " represents the external work performed during the particular process under consideration. In the case of the complete cycle the area of the closed figure on the *p-v* diagram represents the work performed, and the energy change is zero.

Changes of State.—Such changes on the *p-v* plane have various names, depending on the manner in which the volume varies with respect to the pressure: for example, the *isothermal* (constant temperature), the *isobar* (constant pressure), the *constant volume*, and the *adiabatic* (the process during which no heat is added to the substance

nor given up by the substance). The ordinary compression of air or of any other refrigerant during the compression stroke is none of these but is one having the equation pv^n = a constant, where p is the pressure exerted by the gas, v is the volume occupied by the gas in cubic feet, and n is the exponent of the volume, this being usually about 1.4 for air and about 1.28 for ammonia and carbon dioxide. The external work in any of these cases is the projected area on the volume axes of the p-v plane. It is convenient in such cases to apply the energy equation

$$_1Q_2 = U_2 - U_1 + AW$$

in order to find the heat absorbed or rejected. Special formulæ are available in each case and these will be taken up in their turn.

Available Energy.—When heat energy is converted into work, only a part can be so converted, the amount depending on the temperature range in the cycle of operations and on the separate processes. In other words, if heat is supplied to a heat engine, a part, and sometimes only a very small part, is available for, and can be converted into, useful work. The separate processes making up the cycle of operations may be *reversible* or *irreversible*. The theoretical *frictionless adiabatic* and the *isothermal* are examples of the former type, whereas all practical processes are irreversible. By the term "reversibility" is meant that all the events must be taken in the reverse order and that at the end of the process all conditions, both externally and internally, must be restored to their original state.

The Second Law of Thermodynamics and the Carnot Cycle.—The second law of thermodynamics can be stated in a number of ways. From experience, it can be said that:

1. An irreversible change causes a loss of availability.
2. A reversible change causes no change in availability.
3. No increase in available energy in a system can take place as a result of a self-imposed change in the body.

The **Carnot cycle** is one composed of two isothermals and two frictionless adiabatics. If T_1 is the absolute temperature of the source and T_2 is the absolute temperature of the refrigerator, both in absolute degrees, then it may be proved that the efficiency of the Carnot cycle is:

$$E = \frac{T_1 - T_2}{T_1} = \frac{Q_1 - Q_2}{Q_1} = \frac{AW}{Q_1}$$

where Q_1 is the heat supplied from the source, Q_2 is the heat rejected to the refrigerator, and $AW = Q_1 - Q_2$ is the work performed by the engine.

The refrigerating cycle is a reversed cycle (reversed as compared with the steam engine cycle) and it absorbs heat at the lower temperature and rejects it at the upper one, the reversed motor (a compressor) making this cycle a possibility.

Carnot's principle says that such an engine cannot be more efficient than the direct-acting engine working between the same temperature limits. Two examples of Carnot's principle follow:

Problem.—A reversed Carnot engine is used for refrigeration. The temperature of the refrigerator is − 10 deg. F. and that of the hot body is 80 deg. F. Find the horse power required to drive the engine if 10,000 B.t.u. per minute is to be taken from the cold body. The efficiency of the cycle is

$$E = \frac{(80 + 460) - (-10 + 460)}{(80 + 460)} = \frac{90}{540} = 0.1667$$

$$0.1667 = \frac{Q_1 - 10{,}000}{Q_1},$$

and therefore

$$Q_1 = 12{,}000 \text{ B.t.u.}$$

from which

$$AW = 2000 \text{ B.t.u. per minute}$$

$$= 2000 \div 42.44 = 47.1 \text{ hp.}$$

As another example, suppose a reversed Carnot engine is used for heating a building. The outside temperature is 0 deg. F., and the inside of the building is to be maintained at 70 deg. F., and 100,000 B.t.u. per hour are required for heating. Find the minimum horse power required. It seems to be somewhat difficult to conceive such a problem, and to realize that heat can be absorbed from the cold body at 0 deg. F., but similar conditions prevail in every refrigerating plant. The efficiency of the cycle is, as before,

$$E = \frac{T_1 - T_2}{T_1} = \frac{70 - 0}{460 + 70} = 0.132$$

But

$$\frac{Q_1 - Q_2}{Q_1} = 0.132$$

and the problem says that Q_1 is 100,000 B.t.u.;

therefore,

$$\frac{100{,}000 - Q_2}{100{,}000} = 0.132,$$

from which

$$Q_2 = 86{,}800 \text{ B.t.u.}$$

and

$$AW = 13{,}200 \text{ B.t.u. per hour,}$$

and

$$13{,}200 \div 2546 = 5.19 \text{ hp.}$$

Entropy.—In the study of heat processes and cycles it is desirable to have a workable diagram. The diagram on the p-v plane is useful only in a limited manner, as it shows the external work performed and not the manner in which the heat flows. The so-called entropy diagram is one devised to show by the projected area on the entropy axis the amount of heat absorbed or rejected during the process or system. Goodenough[1] defines entropy as a quantity proportional to the unavailable energy. A frictionless adiabatic is one process conceived to have *constant entropy*.

Entropy, being a point function, is expressed by:

$$S_2 - S_1 = \int \frac{dQ}{T} + \int \frac{dH}{T}$$

where $S_2 - S_1$ is the change of entropy, Q is the heat absorbed, H is the heat due to friction, T is the absolute temperature of the substance during the process.

During a frictionless isothermal process

$$S_2 - S_1 = Q \div T \quad \text{and} \quad Q = T(S_2 - S_1) \quad \text{Fig. 1}b$$

Whereas if

$$dQ = Mc_v dt$$

then

$$S_2 - S_1 = Mc_v \int \frac{dt}{T} = Mc_v \log_e \frac{T_2}{T_1}$$

where c_v is a constant.

The temperature-entropy diagram is one with absolute temperature (T) as ordinates and entropy as abscissa. These diagrams are easily drawn for vapors as the entropy in such cases is usually given in the tables of the properties of saturated vapors.

Thermal Potential.—A quantity now being used to a large extent for the solution of thermodynamic problems is one which is called *thermal potential*. This is defined as the sum of the energy and the product Apv, or

$$I = U + Apv$$

As U, p and v are point functions and therefore can be plotted on a set of suitable coordinate axis, it follows that I is a point function and that it can be plotted likewise. Differentiating:

$$di = du + Apdv + Avdp = dq + Avdp$$

Other advantages in the use of I as a coordinate are:

[1] G. A. Goodenough, Principles of Thermodynamics, Henry Holt and Co.

(*a*) If the pressure is constant the heat absorbed (or rejected) is equal to the difference in the value of I at the beginning and end of the process.

(*b*) In a closed cycle the work performed during compression is the difference in the value of I at the beginning and the end of compression.

(*c*) During a throttling process (the process in passing through the expansion valve) the value of I remains constant.

The *P–I* Diagram.—The entropy diagram is useful in understanding the manner of heat flow, but it is not workable to the extent of another diagram called the *P–I* diagram (Fig. 1*d*) or the *I–S* diagram (Fig. 1*c*). The *P–I* diagram is drawn out accurately in Figs. 164, 165, 166, 167 and 168, and these may be used to advantage in the solution of engineering problems.

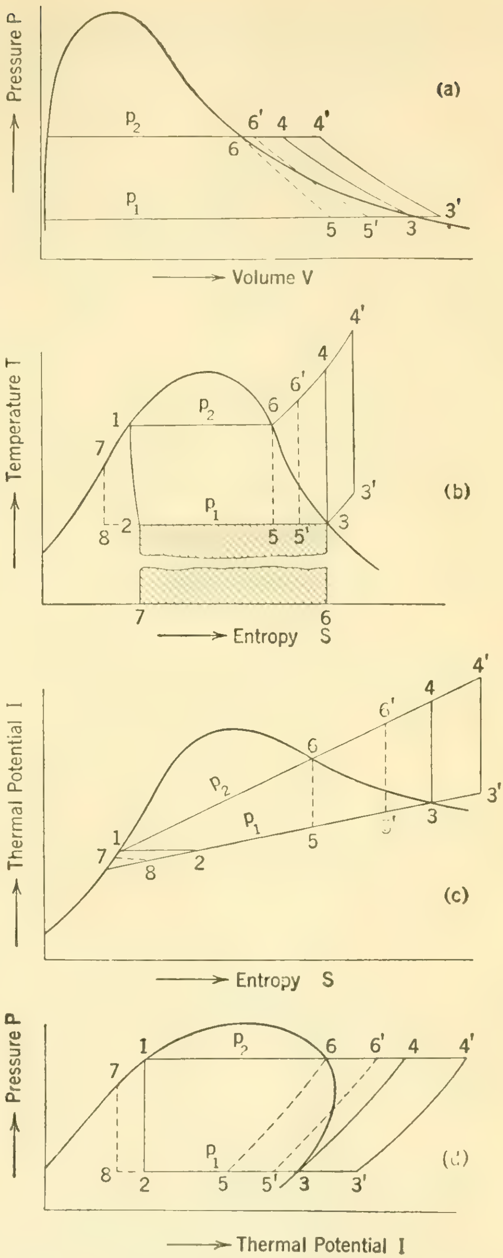


FIG. 1.—Theoretical Refrigerating Cycles for Vapors.

Perfect Gases.—The relationship existing between pressure, temperature and volume for a perfect gas may be expressed by the equation

$$pv = MBT$$

or

$$\frac{pv}{T} = \frac{p_1v_1}{T_1} = \text{a constant}$$

where p is the pressure in pounds per square foot.
v is the volume of the gas in cubic feet.
M is the weight of the gas in pounds.
B is the gas constant, 53.3 for air.
T is the temperature of the gas in degrees F. abs.

Specific Heat of Gases.—The thermal capacity of a substance is the heat required to change by unity one of the variable quantities. The *specific heat* of a substance is the ratio of the thermal capacity to the thermal capacity of water. The usual specific heats of gases are those at constant pressure (c_p) and constant volume (c_v). The ratio of $\frac{c_p}{c_v} = k$ is very nearly constant at 1.4 for diatomic gases, and it is approximately 1.3 for carbon dioxide.

Intrinsic Energy.—The intrinsic energy of a perfect gas has been proved (Joule's Law) to be independent of the volume, and dependent only on the temperature of the gas. This relation is expressed by

$$U_2 - U_1 = Mc_v(t_2 - t_1)$$

where $U_2 - U_1$ = the change of intrinsic energy.
$(t_2 - t_1)$ = change of temperature.

This formula may be written also,

$$U_2 - U_1 = \frac{p_2v_2 - p_1v_1}{k - 1}$$

Changes of State: *Isothermal.*—Should a perfect gas undergo a change of state at constant temperature, then $pv = p_1v_1$ = a constant and the energy change is zero. From the energy equation

$$Q = AW$$

where W is the work performed in foot-pounds. The projected area under the curve on the p-v plane, on the volume axis, gives the external work in every case. For the isothermal the work is given by the expression

$$AW = AMBT_1 \log_e \frac{v_2}{v_1} = Ap_1v_1 \log_e \frac{v_2}{v_1}$$

where v_2 and v_1 represent the final and initial volumes respectively, and e is the base of the Napierian logarithms.

Adiabatic.—During the adiabatic change of state $Q = 0$ and, from the energy equation,

$$U_2 - U_1 = - AW$$

In other words, during the adiabatic change, work is performed at the *expense* of the intrinsic energy or vice versa. The equation for an adiabatic in terms of p and v is given by pv^k = a constant, where $k = \frac{c_p}{c_v}$.

The work performed during such a process is given by

$$_1W_2 = \frac{p_1v_1 - p_2v_2}{k - 1}$$

The relation between the temperatures during an adiabatic process can be shown by

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}$$

The Constant Pressure Process.—During the isobar change of state the heat absorbed is

$$_1Q_2 = Mc_p(t_2 - t_1) = Mc_v(t_2 - t_1) + Ap(v_2 - v_1)$$

The Polytropic Change.—In the polytropic change of state the projected curve on the pv plane can be expressed by the equation

$$pv^n = \text{constant}$$

where n is some exponent which is usually between 1.0 and 1.4 for air The work performed is

$$_1W_2 = \frac{p_1v_1 - p_2v_2}{1 - n}$$

and the energy change (as in any change of state) is

$$U_1 - U_2 = \frac{p_1v_1 - p_2v_2}{k - 1}$$

The Air Refrigerating Machine.—When air or a so-called perfect gas is used for refrigeration it becomes necessary, for economy, to use a compressor and an expansion motor on the same shaft. In such a machine the process for the cooling of commodities will be represented by the line EA (Fig. 2), where A is the temperature of the cold room and F denotes a temperature which is slightly higher than the cooling water

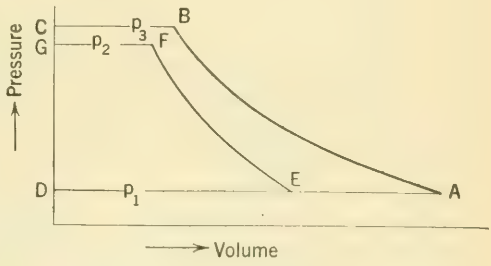


FIG. 2.—Refrigeration Cycle for Air, or a Perfect Gas.

in the aftercooler. The heat absorbed along the line EA, assuming complete expansion in the expansion cylinder, is therefore

$$ {}_EQ_A = Mc_p(t_A - t_E). $$

The net work required to be delivered to the air refrigerating machine (not allowing for friction) is the difference in the area $ABCD$ and $EFGD$. The area $ABCD$ is represented by

$$ W_{\text{compressor}} = \frac{n}{n-1}(p_1v_A - p_3v_B) = \frac{n}{n-1}MB(T_A - T_B) $$

The area $EFGD$ is expressed by

$$ W_{\text{motor}} = \frac{n}{n-1}(p_2v_F - p_1v_E) = \frac{n}{n-1}MB(T_F - T_E) $$

The effect of clearance is to decrease the capacity of the compressor and motor and the result is that the size of both must be increased in proportion. Clearance does not increase the power requirements theoretically but it does practically because of the effect of friction. The formula for the volumetric efficiency is given in Chapter II.

In the formula for the refrigerating effect it is seen that the heat absorbed is directly proportional to the weight M. This weight can be increased per cubic foot of piston displacement by increasing the density of the gas during the suction stroke, and this is the characteristic of the so-called dense air machine.

Vapors.—Practically all refrigerating machines at the present time use volatile liquids, and it is necessary to understand the thermodynamics of liquids and vapors in order to have a clear understanding of the action of the machine. In each of the diagrams mentioned, the p–v, the P–I and the I–S diagrams, there is a liquid line on the left and a dry, saturated line on the right (Fig. 1). The top of the dome made with these two lines is called the *critical temperature.* The region between the liquid and the dry, saturated lines is called the saturated region while the superheated region lies on the extreme right where the vapor begins to show characteristics of a perfect gas.

During the absorption of heat by the liquid the temperature will rise and the volume will increase very slightly. The heat of the liquid (if the specific heat is taken as a constant) is given for one pound of the substance by:

$$ {}_1Q_2 = c \times (t_2 - t_1) $$

and if the specific heat is not constant

$$ dq = cdt $$

where $(t_2 - t_1)$ is the increase in the temperature during the addition of the heat.

If the value of c is not constant then the method of summation must be employed, namely,

$$Q = \int c dt$$

If, at any particular temperature, the action of evaporation takes place heat will be absorbed at constant temperature (and constant pressure) until the liquid is entirely evaporated, and the amount of heat so absorbed per pound is called the *latent heat* of vaporization (r). If after the liquid is entirely evaporated any further heat is absorbed the substance becomes *superheated* and (if the pressure is maintained constant) this occurs at an increasing temperature. The heat absorbed in the superheated region is similar to the action of increasing the temperature of the liquid, and for one pound of the substance, still keeping the pressure constant

$$_1Q_2 = c_p \times (t_2 - t_1)$$

if the specific heat were constant, but as the value varies, then, as before the heat absorbed becomes

$$Q = \int c_p dt$$

There is no standard starting point for the measurement of heat content. For a long time the zero point has been taken as 32 deg. F., but this is awkward in refrigeration so that the Bureau of Standards used -40 deg. F. as their zero in the tables of the properties of saturated and superheated ammonia. At this initial state the thermal potential of the liquid becomes

$$i'_0 = u_0' + Apv_0$$

As $\quad u'_0 = 0$ at this point,

then $\quad i'_0 = Apv_0$

where u_0 and v_0 are the intrinsic energy and volume of one pound of the liquid at the zero of the measurement of heat content.

For example, taking $p = 144 \times 10.41$ lb. per sq. ft.

and $\quad v_0 = \dfrac{1}{43.08},$

then $\quad i'_0 = \dfrac{144 \times 10.41}{778 \times 43.08} = 0.0448$ B.t.u.

The significance of i'_0 can be seen by considering the work performed in forcing one pound of liquid into the container against the pressure p lb.

The value of i is given by the expression $i = i' + xr = i'' - (1.0 - x)r$ for saturated vapor of a quality x, where the term x represents the decimal part (of the unit pound) that is a vapor.

The volume of a mixture is, approximately, $v = xv''$ where v'' is the volume of 1.0 lb. of dry, saturated vapor at the pressure under consideration.

In the tables of the properties of saturated vapors may be found the pressure, temperature, volume of one pound of saturated vapor, thermal potential of the liquid and of the dry saturated vapor, the latent heat and the entropy of the liquid and of saturated gas. If the superheated region is evaluated, the specific volume, the value of I and the entropy are given for the different pressures and amounts of superheat.

The Refrigerating Cycle.—The refrigerating cycle for volatile liquids is shown in Figs. 1*a*, 1*b* and 1*c*. Point 1 represents the liquid under a pressure of p_2 lb. corresponding to some liquefaction temperature usually between 70 and 90 deg. F. In order to perform useful refrigeration the pressure must be reduced to such an amount that the corresponding boiling temperature will be suitable for the conditions prevailing, say, 10 to 20 deg. F. The action of the expansion valve (the pressure reducing valve) is shown by the line *12*. Absorption of heat by the refrigerant takes place along the lines *23* and on the T–S plane the amount of heat absorbed is given by the *area 2367*, but on the I–S diagram the heat absorbed is given by the difference $(i_3 - i_2)$. If the gas with a condition represented by *3* passes to the compressor, superheated gas enters the condenser as the compression line is given by *34*, and the point *4* represents a condition of superheat. If the compression line were represented by the line *56* the condition *5* would indicate the presence of a liquid returning with the vapor, the amount of liquid in this case being sufficient for the point *6* to be just on the dry saturated vapor line. If a greater amount of liquid returned with the vapor then the point *6* might be farther to the left and this would indicate that even after compression there would remain some liquid in the compressed vapor. Finally, the compressed, superheated gas enters the condenser and is cooled and liquefied through the absorption of heat by the condensing water along the line *461*. On the I–S diagram the useful refrigeration is the value $i_3 - i_2$, the work of compression is $i_4 - i_3$ and the heat lost to the condenser water is $i_4 - i_1$. The tables and the diagrams are usually calculated for 1.0 lb. of the refrigerant.

The object of the compressor is simply to increase the pressure as efficiently as possible to an amount sufficiently great so that the refrigerant may be condensed into the condition of a liquid again by the

use of atmospheric air or, in nearly every case, by the use of well or surface water, thereby making it possible to use the same refrigerant continuously. The actual discharge pressure of the compressor, therefore, depends on the temperature and the amount of the cooling water showered over the condenser. The function of the condenser is to liquefy the refrigerant, but it is very important that the pressure of liquefaction be as low as possible as well as the temperature of the liquid at the expansion valve. This can be seen very well from the *I–S* diagram.

The refrigerating effect (from Fig. 1*c*) is seen to be $i_3 - i_2$ B.t.u. per pound, where $i_2 = i_1$, the thermal potential of the liquid at the expansion valve. The lower this value is the greater will be the refrigerating effect $i_3 - i_2$, but the only economical manner of lowering this temperature is by the use of cooling water over the condenser or by the use of a water-cooled aftercooler. During the compression process, if the compression line can be represented by the expression pv^n = a constant, the temperature rise may be excessive and is given by the formula

$$t_2 = (t_1 + 460)\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 460$$

where t_2 = the final discharge temperature in deg. F., t_1 = the initial temperature in deg. F., and p_2 and p_1 are the final and the initial pressures respectively.

If recourse is made to the water jacket only in order to keep from overheating the cylinder the temperature of the discharge will become high at times, sometimes as great as 300 deg. F. At one time it was generally understood that improved operation could be obtained by the use of a liquid refrigerant injection into the suction of the compressor and such a process has the term *wet compression.* Such a means of operation is still used in extreme conditions but the control is difficult and few engineers advocate wet compression at the present time for general practice.

The Ton of Refrigeration.—The American ton of refrigeration is the cooling rate of 200 B.t.u. per minute or 12,000 B.t.u. per hour. The standard operating conditions are an evaporation temperature in the cooling coils of 5 deg. F. and a liquefaction temperature in the condenser of 86 degrees. The unit of refrigeration in British practice is the cooling effect of one kilogram calorie per second. This would be $3.96 \times 60 = 237.6$ B.t.u. per minute or about 18.8 per cent greater than the American unit. The American ton of refrigeration got its origin from the consideration of the melting effect of 2000 lb. of ice in

24 hr. with the latent heat of fusion, taken in round numbers, as 144 B.t.u. per pound.

The Coefficient of Performance.—According to the laws of thermodynamics the work performed in compressing different refrigerants for the same heat absorbed from the refrigerator is the same (this being another way of expressing the second law) for similar sets of conditions. In practice there are variations which assume relative proportions of as much as 20 per cent for carbon dioxide as compared with ammonia. This variation is due to the relatively large value of the heat of the liquid as compared with the latent heat of vaporization and because a pressure reducing valve is used instead of an expansion cylinder. It is important that the power consumed be known for the different refrigerants under varying operating conditions. This amount can be found, among other ways, from the ratio of the refrigerating effect to the work of compression. On the *I–S* diagram, Fig. 1*c*, the ratio would be $\dfrac{i_3 - i_2}{i_4 - i_3}$. For efficient work the value of the coefficient of performance should be large, with theoretical values of from 4 to 10, but in practice the coefficient is considerably less.

Horse Power per Ton of Refrigeration.—In American practice it is more usual to hear the term horse power per ton of refrigeration than the coefficient of performance. Using again the *I–S* diagram such a value would be obtained theoretically from

$$\text{horse power per ton} = \frac{i_4 - i_3}{i_3 - i_2} \times \frac{200}{42.44} = 4.713\,\frac{i_4 - i_3}{i_3 - i_2}$$

In practical work the real volumetric efficiency would need to be considered, and this would increase the values given in the formula. Values for horse power per ton of refrigeration may be less than 1.0 up to 2.0 or more depending on the operating conditions.

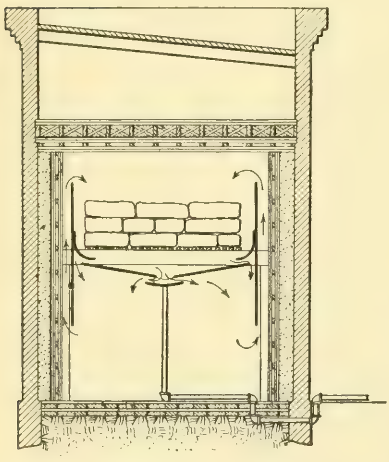


FIG. 3.—Refrigeration by the Use of Ice.

Refrigeration.—Refrigeration, as will be noted later, involves a multitude of different applications. In general it is understood to embrace any method of securing a temperature lower than that of the atmosphere. If this is done by means of ice and salt, or other mix-

tures, or by the use of ice alone (Fig. 3) the work is limited and expensive. The temperatures produced by the use of various mixtures are as follows:

TABLE 1

	Parts by Weight	Degrees F.		Parts by Weight	Degrees F.
Sodium chloride	1	−1.6	Sodium chloride	1	−0.4
Snow (or ice)	3		Snow	1	
Calcium chloride	1½	−27.4	Calcium chloride	2	−43.6
Snow	1		Snow	1	
Dilute nitric acid	1	−31.0	Saltpeter	1	
Snow	1		Sal ammoniac	1	−11.2
			Water	1	
Potassium hydrate	4	−38.3	Sulphuric acid		
Snow	3		Nitric acid	1	−40.0
			Snow	2	

Such a method of securing refrigeration is cumbersome, limited as to the temperatures secured and the ability to secure ice and the cost of producing the cooling would be too great for general use. If the cooling is done by means of a machine of any sort, the operation is called *mechanical refrigeration.*

The Simple Refrigerating Cycle.—The modern refrigerating plant, Fig. 4, uses a closed cycle, even when air[2] is used as the refrigerant. The complete cycle includes then: a set of refrigerating coils through which the refrigerant circulates and absorbs heat from the cold body; an aftercooler or condenser (at the upper pressure) which is the hot body and which receives heat; and a reversed engine, Fig. 5, usually called the compressor. Between the condenser and the refrigerating coils is the throttle valve for reducing the pressure to the amount best suited for the conditions in the refrigerating coils. A well-designed plant must have the work of each of these three parts of the cycle well

[2] A closed cycle makes it possible to operate with suction pressures above one atmosphere as in the case of the dense air machine. With such closed cycles the moisture in the air may be removed so as to decrease the operating troubles due to frost accumulations on the expansion engine valves and exhaust ports. It so happens that in the early days of mechanical refrigeration the open system was used more extensively than the closed cycle.

balanced. The refrigerating coils must be large enough in heat transfer surface to absorb the required amount of heat from the substance to be cooled, for example, air, liquids, etc., in order that the required " duty " shall be performed. The absorption of heat involves consideration of the value of heat transfer usually stated as the number of B.t.u. per

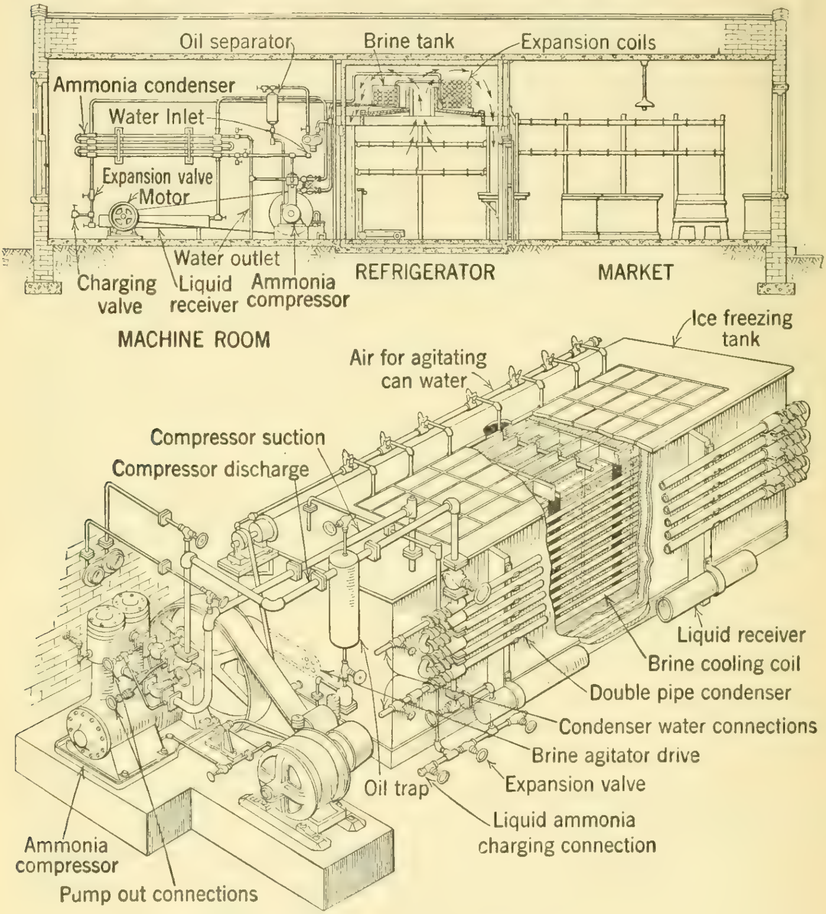
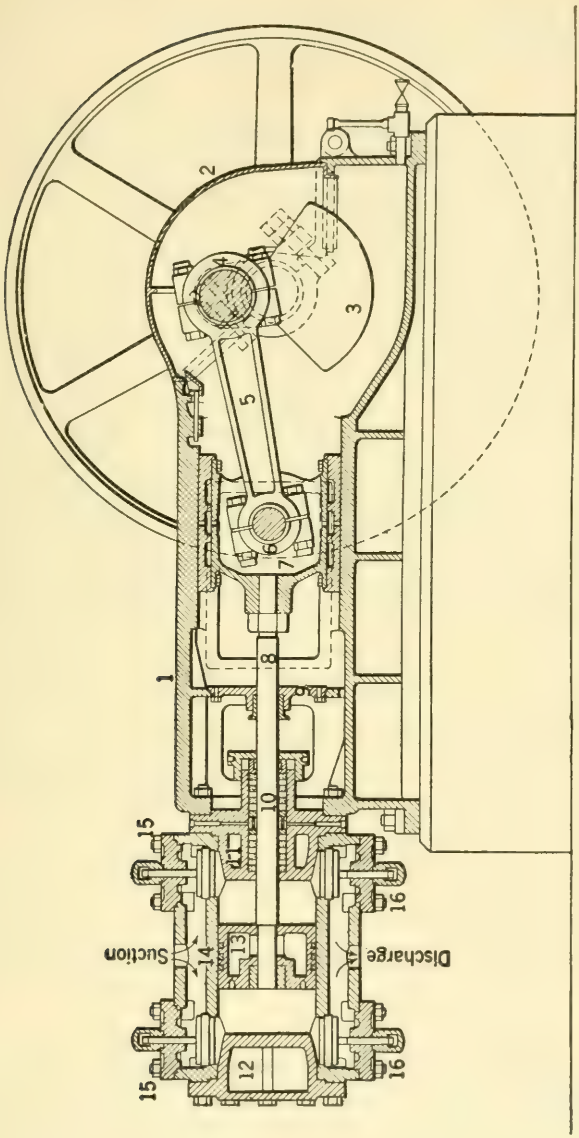


FIG. 4.—Typical Refrigeration Plants.

square foot per 1.0 degree F. difference of temperature per hour and is represented by the symbol k. This value k is of extreme importance in mechanical refrigeration.

The aftercooler (the condenser) has to remove a certain amount of heat, an amount varying from 230 to 250 B.t.u. per minute per ton of



SECTIONAL ILLUSTRATION

1. Frame
2. Crankcase Cover
3. Counter Balanced Crankshaft
4. Crankpin Bearing
5. Connecting Rod
6. Crosshead Pin Bearing
7. Crosshead
8. Piston Rod
9. Oil Guard with Wiper Rings
10. Stuffing Box for Piston Rod
11. Front Cylinder Head
12. Back Cylinder Head
13. Piston
14. Cylinder
15. Suction Valve
16. Discharge Valve

FIG. 5.—Typical Horizontal Double Acting Ammonia Compression Cylinder.

refrigeration. Again the value of the coefficient of heat transfer (k) is an important consideration and in each case the total heat transfer may be stated from the formula:

$$Q = A \times k \times (t_2 - t_1)$$

where A = the area of the surface exposed to heat transfer;
$t_2 - t_1$ = the temperature difference on the two sides of the surface;
Q = the heat absorbed or rejected per hour.

Expansion Cylinder or Expansion Valve.—As the energy change of a perfect gas is a function of the temperature only it becomes essential that when a so-called perfect gas is used for refrigeration that an expansion cylinder be employed in order that on expansion it will at least approximate the adiabatic. In the case of vapors this is different and in practice all successful refrigerating machines using a liquid refrigerant make use of the throttle (pressure reducing) or the so-called "expansion" valve. The gain in simplicity, cheapness in first cost and ease in operation as compared with some slight gain in economy when using the expansion cylinder has entirely excluded the latter. As an example, take a condition of 86 deg. F. temperature of the liquid ammonia and 5 deg. F. evaporating temperature. In the expansion valve $i_1 = i_2$ or

$$i'_{86} = i'_5 - x_5 r_5 \quad \text{or} \quad x = \frac{138.9 - 48.3}{565.0} = 0.1605$$

whereas in the adiabatic expansion (constant entropy)

$$s'_{86} = s'_5 + x_2\left(\frac{r}{T}\right)_5 \qquad x = \frac{0.2875 - 0.1092}{1.2161} = 0.1470$$

From this calculation the loss of refrigeration due to the use of the expansion valve is (0.1605 − 0.1470) 565 = 7.6 B.t.u.

Sub-cooling the Liquid.—The refrigerating effect, per pound of ammonia, boiled to dry, saturated vapor with t_1 before and t_2 after the expansion value is

$$Q = i''_2 - i'_1$$

It is evident that any scheme which will decrease the value of i'_1 (the thermal potential of the liquid before it enters the expansion valve) will result in a greater value of the net available refrigeration per pound of refrigerant, but in general the operation must be done by the condensing water. However if colder water than that used for liquefaction in the condenser is available it must be used for sub-cooling *after* liquefaction as, if any vapor is present sub-cooling is not possible unless air or other non-condensible gases are present as is shown in Chapter IV.

The only other successful method of sub-cooling the liquid is to make use of the dual compression or the stage compression method given in Chapter II whereby the liquid is cooled at an intermediate pressure and the gas evolved at the intermediate pressure is compressed by a special compressor.

The Optimum Refrigerant.—The second law of thermodynamics states that the kind of working medium does not influence the economy of operation of a heat engine. A preferred refrigerant, therefore, will not be one that will increase the efficiency of the cycle, but it will be one that for the kind of operation, or the type of load or service will give best results as regards the pressures exerted in the cycle, the piston displacement, the chemical properties or the effects on plant or animal life. Some of the small machines are required, by reasons of convenience, to use an air-cooled condenser in which case the pressure resulting from the liquefaction must not be excessive.

In the following chapters of this handbook, then, there will be considered first the compressor or other mechanical device between the lower (the refrigerating) pressure and the higher (the condenser) pressure, Chapter II; the absorption machine, Chapter III; condensers, fittings, etc., Chapter IV; the low-pressure side and its applications to various industries; as for example that of ice making, Chapter XIII; cold storage, Chapter XIV; and miscellaneous applications, Chapter XV. Subjects of lesser general importance, though valuable to the engineer, such as heat transfer, Chapter VI; automatic refrigeration, Chapter V; brine, Chapter VIII; water-supply systems, Chapter IX; operation and erection, Chapter X, etc., are discussed as indicated.

CHAPTER II

THE COMPRESSOR

History and Development.—The first machines used in the mechanical production of cold were constructed on the principle of the vacuum machine, wherein the vacuum—which was obtained mechanically—permitted the refrigerant to boil at a sufficiently low temperature to secure the results desired. These machines included that of Wm. Cullen in 1755, using water under high vacuum; Vallance's machine of 1824; and Edmund Carré's device of 1850, both using sulphuric acid to absorb the water vapor, and thereby permitting less severe operating conditions than would be required if an air pump had had to be used.

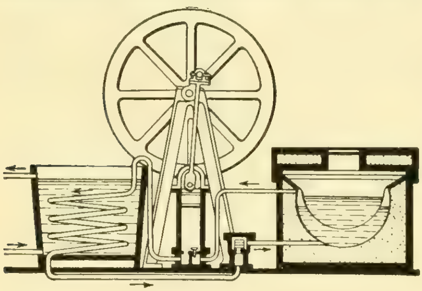


FIG. 6.—Vapor Compression Refrigerating Machine of Jacob Perkins.

In 1845 Dr. John Gorrie developed the cold air machine, using a closed cycle and an expansion cylinder, and this machine was improved by Kirk in 1861, Postle in 1868 and Windhausen and Nehrlich in 1869. The air machine was perfected by James Coleman and John and Henry Bell in 1877 and subsequent years, with the result that the Bell-Coleman compressor became very well known and mechanical refrigeration got a real impetus. The present-day compressor is based on the designs of Linde, Ferguson and John De la Vergne.

The first refrigerant other than air or water was that used by Jacob Perkins whose compression machine (Fig. 6) was invented in 1834. This machine was intended to be used with sulphuric ether, $[(C_2H_5)_2O]$ but it never got beyond the experimental stage. The Perkins machine was revived in 1857 by Dr. James Harrison (using sulphuric ether) and was quite successful for refrigeration in breweries, and for the cooling of meat and other perishable products. Finally Dr. Carl Linde, in 1873, introduced the ammonia compression machine, and in 1876

Raoul Pictet devised the sulphur dioxide compressor, using water to cool the piston and piston rod as well as the cylinder. In the eighties mixtures[1] of different refrigerants, and volatile liquids like ethyl chloride, methyl chloride, ethylic ether, acetylene, ethylo-sulphurous dioxide, naphtha, gasoline, cyrogene, etc., were given attention. By a system of elimination the practical refrigerants were narrowed—as far as Great Britain and the United States were concerned—to ammonia and carbon dioxide, but at the present time (1927) interest is directed towards other refrigerants for special purposes.

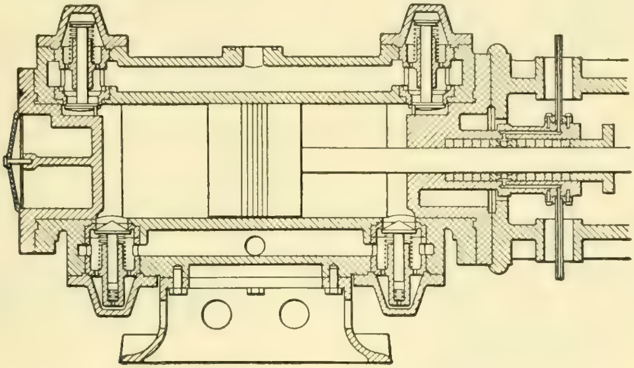


FIG. 7.—Apeldoorn Compressor.

THE MODERN DESIGN OF AMMONIA COMPRESSORS

The Slow-speed Horizontal Double-acting Compressor.—The present-day construction of the older type (the Linde design) of ammonia compressor of the double-acting, horizontal design (which is still preferred by certain engineers) is heavy and sturdy. The design is more like the heavy-duty rolling-mill engine, and has sufficient metal in the cylinder for several reborings. The frames are usually made of very close-grained cast iron, while the cylinder is either of a close-grained cast iron or of a special steel casting and is cast separately from the frame. The cylinder heads are made of the same material as the cylinder, and the Linde type of compressor has a spherical piston and

[1] A number of mixtures of two vapors has been attempted of which Pictet's liquid is the best known. This latter is a certain mixture of carbon dioxide and sulphur dioxide and the resulting combination condenses at a pressure considerably less than required for pure carbon dioxide. More recently propane and butane have been proposed but they are mere mixtures and not new compounds. The physical properties of such *mixtures* are never known with any degree of accuracy, and the relative amounts of the mixture vary considerably unless it is possible to eliminate all leaks.

cylinder heads, the idea being that greater valve area as well as greater strength is possible by this arrangement.

In this type the piston is now, usually, of semi-steel, cast in one piece and hollow, although the built-up construction using the spider, spring and bull ring is also common. The crosshead is usually of the box type with adjustable shoes at top and bottom, and is of semi-steel. The connecting rod is made of one-piece steel forging, with bronze boxes, lined with babbitt. The main bearing is of the floating type of quarter box, lined with babbitt, and is adjusted by means of wedges and screws (Fig. 8). The crankshaft is of one-piece steel forging, turned and ground. The stuffing box is of the lantern type, using now, as a rule, first a metallic packing; then the lantern, and finally, a soft packing. The lantern is connected by means of piping to the suction line so that there is in this way only suction pressure exerted on the outer part of the stuffing box, whose length may vary in the different makes from 8 to 18 in. (Fig. 9).

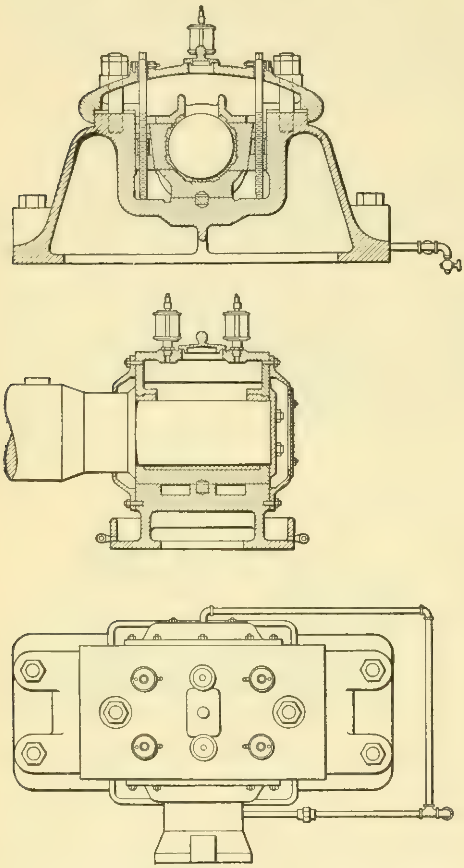


Fig. 8.—Compressor Details; the Main Bearing.

The older Linde type of valve is the poppet, or mushroom valve which gives "line contact" on a bevel seat. The poppet valve has been remarkably successful. Springs are used to close the valves, but an attempt is made to cushion the closing of the valve (which is usually quite heavy) by means of dash pots (Fig. 10). On account of the sturdy design and workable valves the compressors of the Linde type have had a very successful life, the design not having been changed to any great extent since the first construction in about the year 1873.

The poppet valve can be made very tight, and it can be reground on its seat easily. The compressor speed is limited, however, to from 75 to 100 r.p.m., and therefore where higher speeds are required a lighter weight valve—one of small inertia—must be used.

The Horizontal High-speed Compressor.—The principal reason for the use of the high-speed in place of the low-speed compressor is the decrease in the space required by the machine, although a cheaper electric motor can be used if it is permitted to revolve at 1200 or 1800 r.p.m., and the oil engine drive is easier to make at compressor speeds up to 250 r.p.m. It is a mistake to say, however, that the moderate-speed compressor is cheaper per ton of refrigeration, as the stresses developed make it necessary to use higher-priced material than would

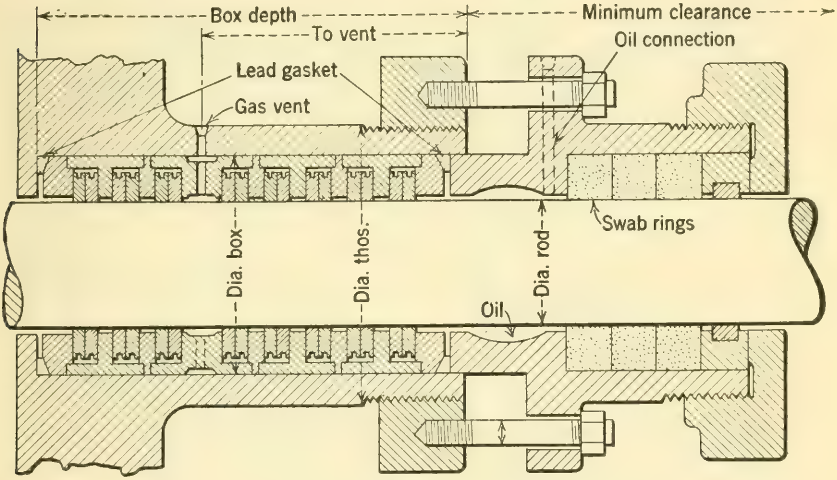


Fig. 9.—Metallic Rod Packing.

be used in the slow-speed machine, and the problems of balancing, lubrication and foundations demand special consideration, in consequence of which the cost is considerably increased, as compared with the slower rotative speeds using the same sized cylinders.

The high-speed compressor frame is of the rolling-mill engine type, and the frame, cylinder and heads are composed of semi-steel (Fig. 11). The castings are sand blasted and then have an acid bath to remove the surface sand. The piston and connecting rods and the crankshaft are of high quality steel forgings, each forging being made from a single billet. The pins are of high carbon steel, heat treated and ground. The valves are of the plate (the De la Vergne, and Ingersoll-Rand), Fig. (11) the ring plate (Vilter, and discharge valve of the Arctic) and

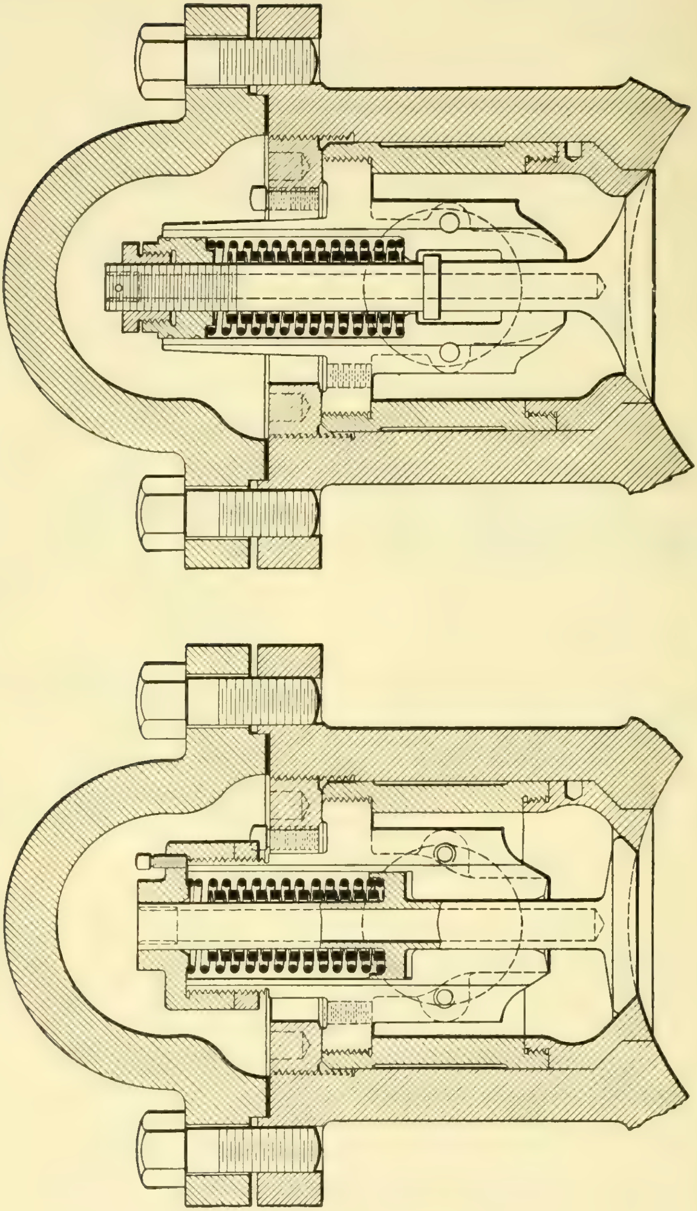


FIG. 10.—Compressor Details; Poppet Valves.

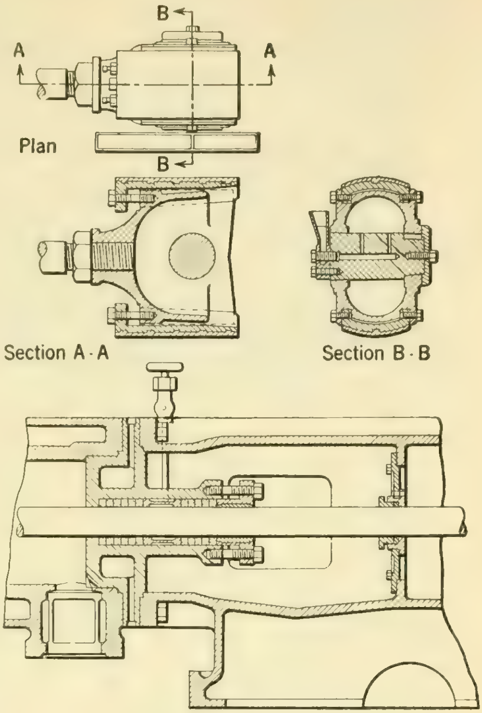


FIG. 11a.—Compressor Details; Crosshead and Rod.

the ribbon (the Carbondale) type. The material composing the valves is the very best, frequently chrome-vanadium steel, and is hardened and ground, or tempered and ground. The rise of the valve off its seat is very slight. Typical indicator cards are shown in Figs. 13 and 14.

In the horizontal machines the discharge valves are always placed on the bottom in order to permit easy relief should liquid ammonia enter the cylinder with the suction gas. The high-speed compressor uses sets of valves of small size in preference to a smaller number of large valves. The suction and the discharge valves are frequently identical, except for the position of the valve cage in the seat. The ports are

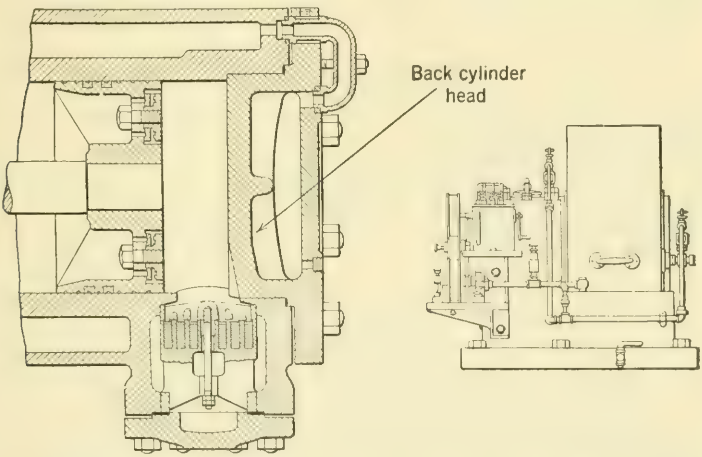


FIG. 11b.—Compressor Details; Head End and Valves.

made generous in size and as direct as possible. Tests indicate that the port design affects the economical operation to a greater degree

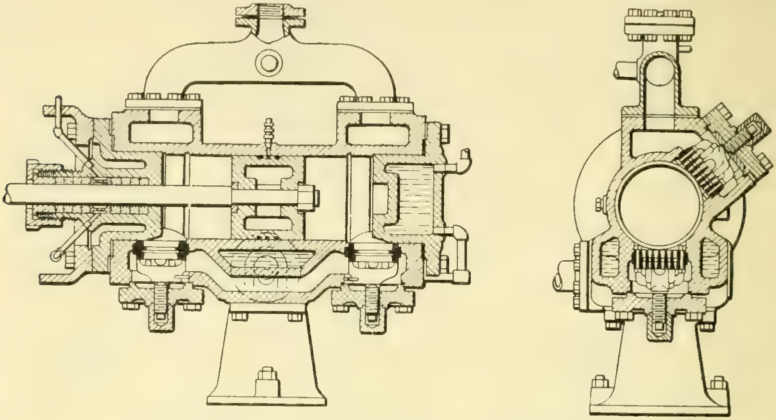


FIG. 12.—Horizontal Double-acting Ammonia Compressor with Plate Valves.

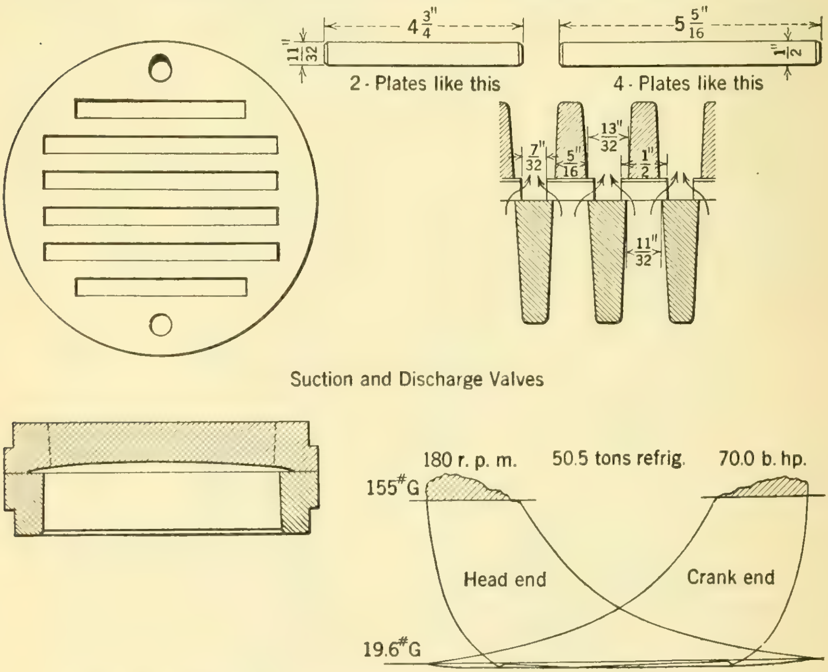


FIG. 13.—Details of the Ribbon Type of Valve, with Indicator Card.

than the net cross section of the valve opening. Since designers act with the understanding that a nominal amount of clearance—varying

from 3 to 6 per cent—does not affect appreciably the economical operation of refrigerating compressors, the horizontal, double-acting high-speed compressor usually does not attempt to eliminate clearance, or attempt to utilize the safety head principle, nor is there an attempt to safeguard the cylinder, by flexible cylinder head studs or otherwise, from the harmful effects of slugs of ammonia liquid in the cylinder.

Special consideration, however, is given to *lubrication.* There are now two methods in frequent use; first, the sight feed automatic lubricator (the Vilter and the De la Vergne) and second, the splash type for the engine crank mechanism (the Ingersoll-Rand, the Arctic and the Worthington). When the latter is used, provision is required in the shape of a special partition to prevent the oil in the crankshaft from entering the cylinder. Cylinder (low temperature test) oil is pumped into the cylinder and the stuffing box by means of a special mechanical pump lubricator.

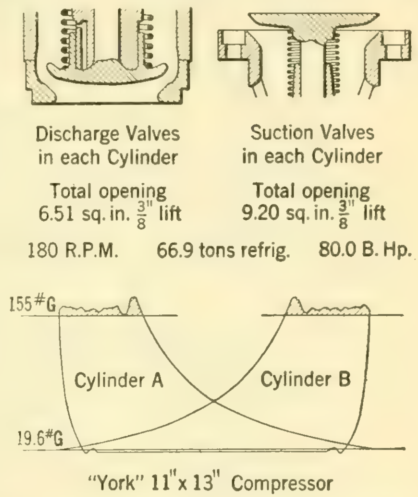
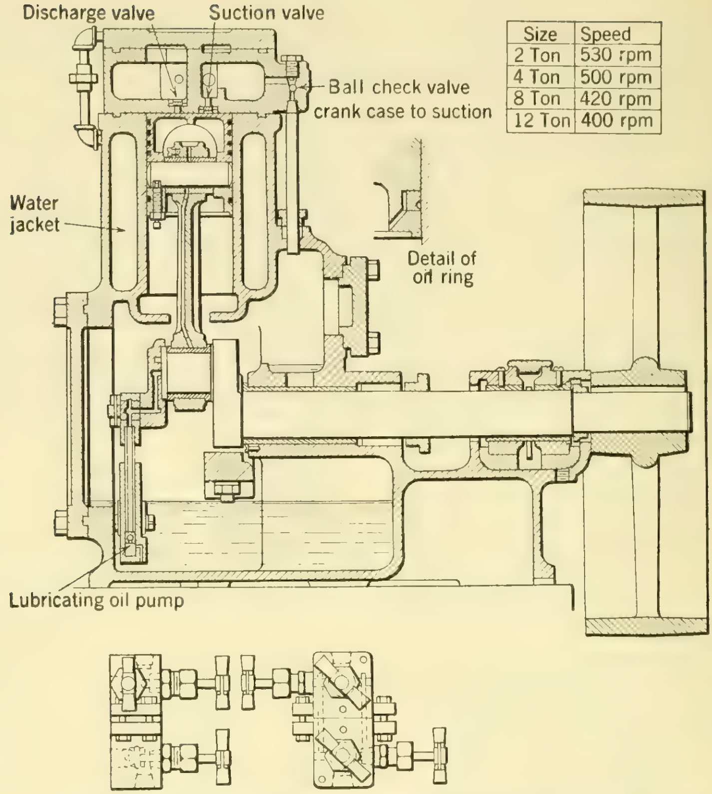


Fig. 14.—Details of the Poppet Type of Valve, with Indicator Diagram.

As it is practically impossible so to control wet compression as to obtain the best results, all modern compressors operate on dry compression, thus necessitating (for ammonia) the use of a water jacket. The water jacket is of slight use except at the last third of the stroke, and so the heads are practically the only place where the horizontal, double-acting compressor can obtain any benefit. This tendency is seen in the Vilter, the Ingersoll-Rand and the Worthington Pump designs (Figs. 11, 12 and 24). It is a question whether the water jacket affects the compression line appreciably, but its function is to prevent overheating of the metal, and in consequence to improve at the same time the " real " volumetric efficiency.

The Vertical Single-acting Compressor.—The vertical single-acting compressor is of two types, (1) the semi-enclosed type and (2) the totally enclosed designs. The semi-enclosed twin cylinder open frame compressor first appeared about 1900 and is still being sold by the Frick and the York Companies (Fig. 26). It is an excellent design, giving the greatest economy of any compressor, and is built in the



Size	Speed
2 Ton	530 rpm
4 Ton	500 rpm
8 Ton	420 rpm
12 Ton	400 rpm

FIG. 15.—Single-acting, Enclosed Type, High-speed Ammonia Compressor.

largest capacities. Because of the vertical cylinder and the use of a crosshead, only piston ring pressure is exerted on the cylinder walls, and the cylinder does not wear out of round as quickly as the horizontal design does. The height of the compressors, the operating troubles with the stuffing box and the relatively slow speed appear to influence opinion in favor of the totally enclosed compressor (Fig. 16). This latter design is becoming more and more liked, both in America and Great Britain. Both types have a uni-flow of the ammonia as the suction ammonia gas enters below the piston and passes through the suction

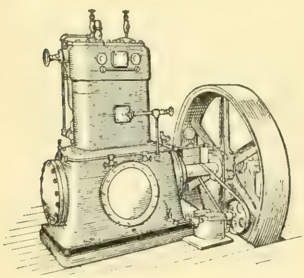


FIG. 16.—Twin Vertical Single-acting Ammonia Compressor.

valve in the piston of the ring plate or the balanced light weight poppet type, the latter opening and closing by the inertia of the valve during reciprocation, and the discharge passing through a form of poppet valve in the false head held on its seat by means of heavy springs (Fig. 14). The water jacket is located along the *last third* of the stroke only, and at times in the head as well, the remainder of the barrel being insulated. On account of the safety false head the (striking) clearance

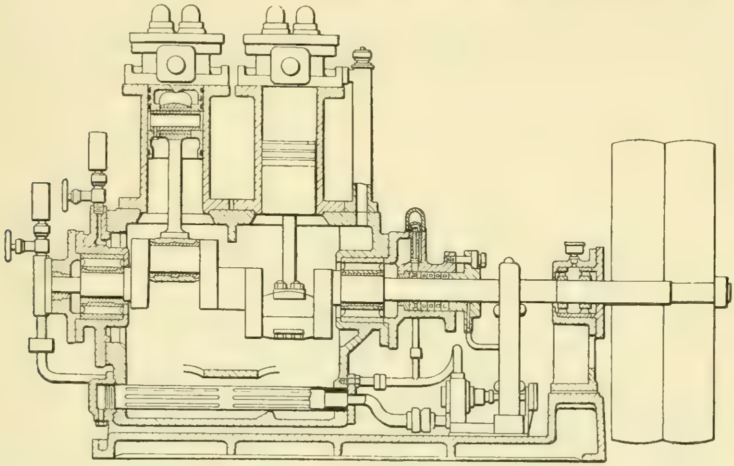


FIG. 17.—The Haslam Ammonia Compressor.

From The Haslam Foundry & Engineering Co., Ltd., Derby, England.

	Number of Machine				
	2	3	4	5	6
Heat units eliminated hourly, cooling 60° to 40° F.	32,000	72,000	112,000	152,000	240,000
Ice-making capacity per 24 hours, tons	1	$2\frac{1}{3}$	$3\frac{2}{3}$	5	8
Refrigerating capacity per 24 hours, tons	2	$4\frac{1}{2}$	7	$9\frac{1}{2}$	15
Actual horse power required	$4\frac{1}{2}$	9	13	16	22
Revolutions per minute	200	170	150	140	130
Shipping weight, cwts.	$13\frac{3}{4}$	$19\frac{1}{2}$	$25\frac{1}{2}$	$36\frac{1}{2}$	$48\frac{1}{2}$
Shipping measurements, cubic feet	36	75	100	170	220

is made very small, as little as $\frac{1}{64}$ in. The piston rods and stuffing box are subjected to suction pressure only.

The Semi-enclosed Compressor.—The semi-enclosed vertical single-acting compressor (Fig. 26) has its two cranks 180 degrees apart, and (if engine driven) has the connecting rod of the steam engine connected to one of the crankpins of the compressor. This permits the maximum resistance of the compressor to occur at the same time as the maximum

turning effort of the horizontal engine, and thereby permits a smaller flywheel than would otherwise be needed. A regular crosshead is used, but the stuffing box, being subjected to suction pressure only, need not be as long as is required for the double-acting machine. This stuffing-box packing has hard usage, and as it is very hard to keep the operating conditions constant, and as the packing is likely to become frozen from time to time when liquid ammonia returns with the suction back to the compressor, the liquid is likely to run down the rod. The materials used in the cylinders are close-grained, high-grade, cast iron, but the frame and the remainder of the machine are usually of ordinary material.

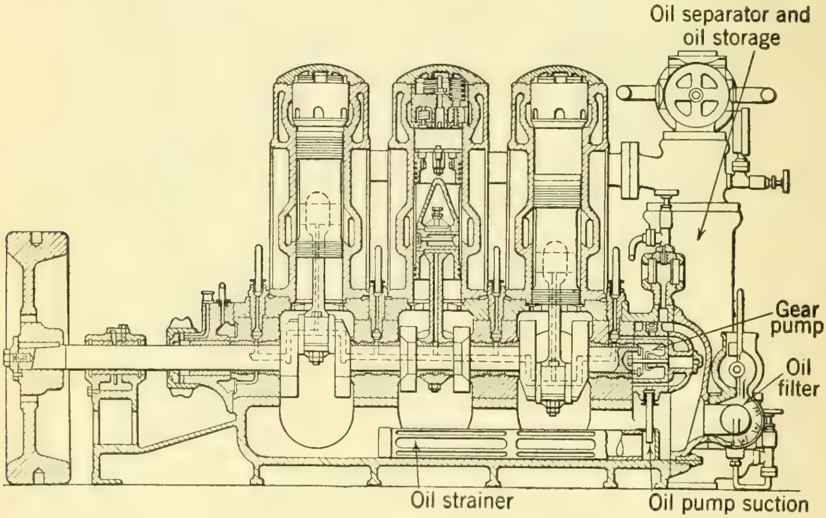


Fig. 18.—The Stearns Ammonia Compressor.

The Enclosed Compressor.—The enclosed crankcase compressor (Figs. 15, 16, 18) lends itself best to high speed, or moderate speeds in the larger sizes, to automatic operation and to decreased loss of ammonia. It has been popular in the smaller sizes, say from the $3'' \times 3''$ to the $9'' \times 9''$ but now the enclosed compressor is found in sizes up to $18'' \times 20''$. The feature tending to decrease the loss of ammonia is the fact that the packing and the rod are kept at nearly constant temperature and that the rod has rotation and not translation. A scored rod need not leak under conditions of rotation.

The cylinder construction as regards materials, heat treatment and machine work is practically the same as for the high-speed horizontal compressor. The cylinder is of semi-steel and the crankcase and the

cylinder are sand-blasted and acid-dipped in order to remove any surface sand. The piston is known as the double trunk pattern as it acts in the capacity of a crosshead, and takes the side thrust due to the obliquity of the connecting rod, as well as performing the ordinary function of the piston. The piston in the smaller sizes, up to the 9″ × 9″ size, usually has only one balanced suction valve in the piston, but the larger sizes have three or more such valves. The discharge valves, usually of the modified plate or the poppet type of valve, may be three or more in number.

Unusual care is necessary in the design and the construction of this type of compressor. It is subjected to very heavy forces. The machine work in the cylinder and the crank-case must be excellent, and heat treatment usually is necessary in order to prevent warping of the casting after the finishing cuts are made, a consideration that is even more important where three or more cylinders are used (Fig. 19). The crankshaft needs to be very massive and of the best material, particularly when wear on the part of the center bearings means a much longer span than the difference between cylinders. The York Manufacturing Company specified open hearth steel forgings of 75,000 lb. per square inch ultimate tensile strength, and an elastic limit of 40,000 lb. per square inch for both crankshaft and connecting rods.

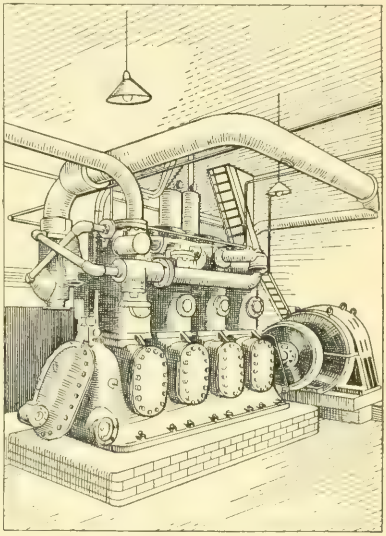


FIG. 19.—The Stearns Compressor.

Clearance.—The feeling has prevailed for years that clearance, in an ammonia compressor particularly, is a very bad thing, and for some time past an attempt has been made to reduce the clearance as much as possible. It is now understood that clearance will reduce the capacity of the refrigerating machine, since the gas in the clearance volume will expand at the beginning of the suction stroke until the pressure, having finally been reduced in the cylinder to an amount equal to or less than the pressure in the suction bends, will cause the gas to enter the cylinder and the new *effective* suction stroke will begin. The work performed in the compression of the gas in the clearance volume is very nearly all returned

in the expansion taking place at the beginning of the suction stroke. The loss of capacity, however, is a real one and may reach a large amount, depending on the range of the pressure and the clearance in the cylinder in per cent of the piston displacement. It so happens that certain forms of valves make it necessary to use some 4 or more per cent of clearance, but in such cases the usual procedure is to increase the diameter slightly in order to compensate for the loss of capacity.

Volumetric Efficiency.—In refrigeration, when the volumetric efficiency is spoken of, reference is not being made to the *apparent*[2] volu-

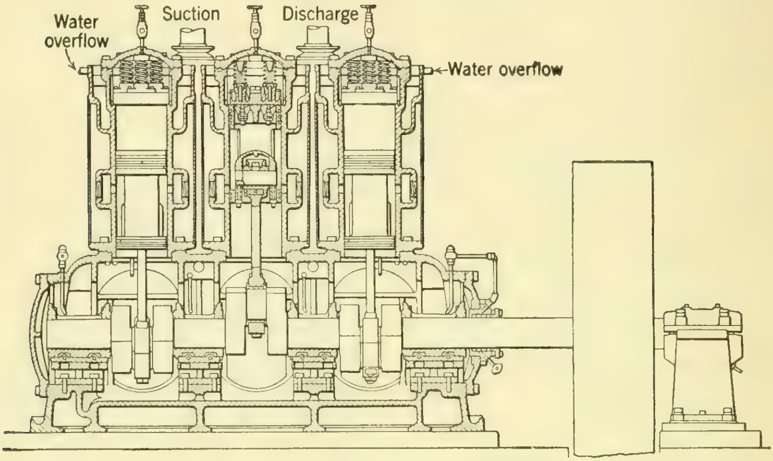


FIG. 20.—The York Ammonia Compressor.

metric efficiency which is caused by the clearance, but to the *real* volumetric efficiency. This is in reality a quantity obtained by dividing

[2] The apparent volumetric efficiency may be found from the following values, using the quantities given in Fig. 21.

$$V_1 = V_p + V_c - V_d$$

$$V_d = V_c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} \quad \text{and let} \quad \frac{V_c}{V_p} = m$$

$$V_1 = V_p + mV_p - mV_p\left[\frac{p_2}{p_1}\right]^{\frac{1}{n}} = V_p\left[1 + m - m\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\right]$$

The volumetric efficiency is the ratio $V_1 \div V_p = 1 + m - m\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$ where n for an adiabatic process with ammonia is about 1.3, but for usual expansion and compression it is better to take the value as 1.28.

the actual weight of the gas pumped multiplied by the specific volume of the saturated gas at the suction pressure by the piston displacement. The real volumetric efficiency takes into account the superheating of the suction gas due to the effects of the cylinder walls, piston, valves, etc., which are always at some temperature intermediate between the temperature of the discharge and the suction gas. This effect of the cylinder walls cannot be shown on the indicator diagram. Its value for ammonia and carbon dioxide is shown in Figs. 22 and 23. The superheating of the suction gas is an absolute loss, both as regards the work done on the gas and the capacity of the compressor. The uniflow principle of compressor design, and the use of stage compression are the best means of reducing this loss.

Speed of the Compressor.—For considerable time the piston speed has been limited to about 400 ft. per minute, but this limitation has not been general, since, for example, in air compression a speed of 600 ft. per minute has been used with success, and steam and gas engine practice has permitted 800 ft. per minute. The limit of the piston speed had been nothing more than the ability to lubricate properly both the piston and the cylinder walls.

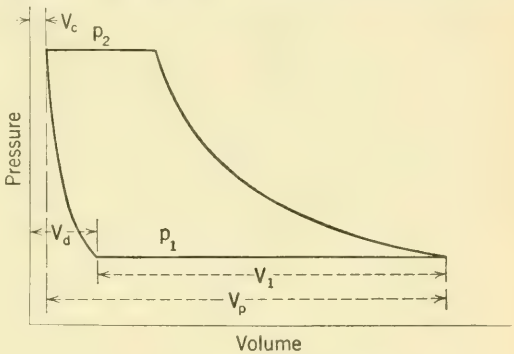


FIG. 21.—Indicator Card.

The rotative speed, however, is another matter, as the number of strokes determines the number of times the valves have to open and close per minute. In order to increase the rotative speed the compressor needs to have a light weight valve and one that has a liberal net opening for the passage of the gas. As the number of revolutions per minute increases, the amount of the valve lift has to diminish, or pounding of the valve will be the result. The outcome of these factors has been the almost universal use of the plate, the ring plate or the ribbon valve. With the exception of the sleeve valve compressor (Fig. 47), mechanically operated valves are never used in refrigeration, but two types of "inertia" valves are used: the balanced piston valve used in the vertical single-acting compressor, and the special suction valve of the ring-plate type used by the Arctic Ice Machine Company in their horizontal double-single-acting compressors.

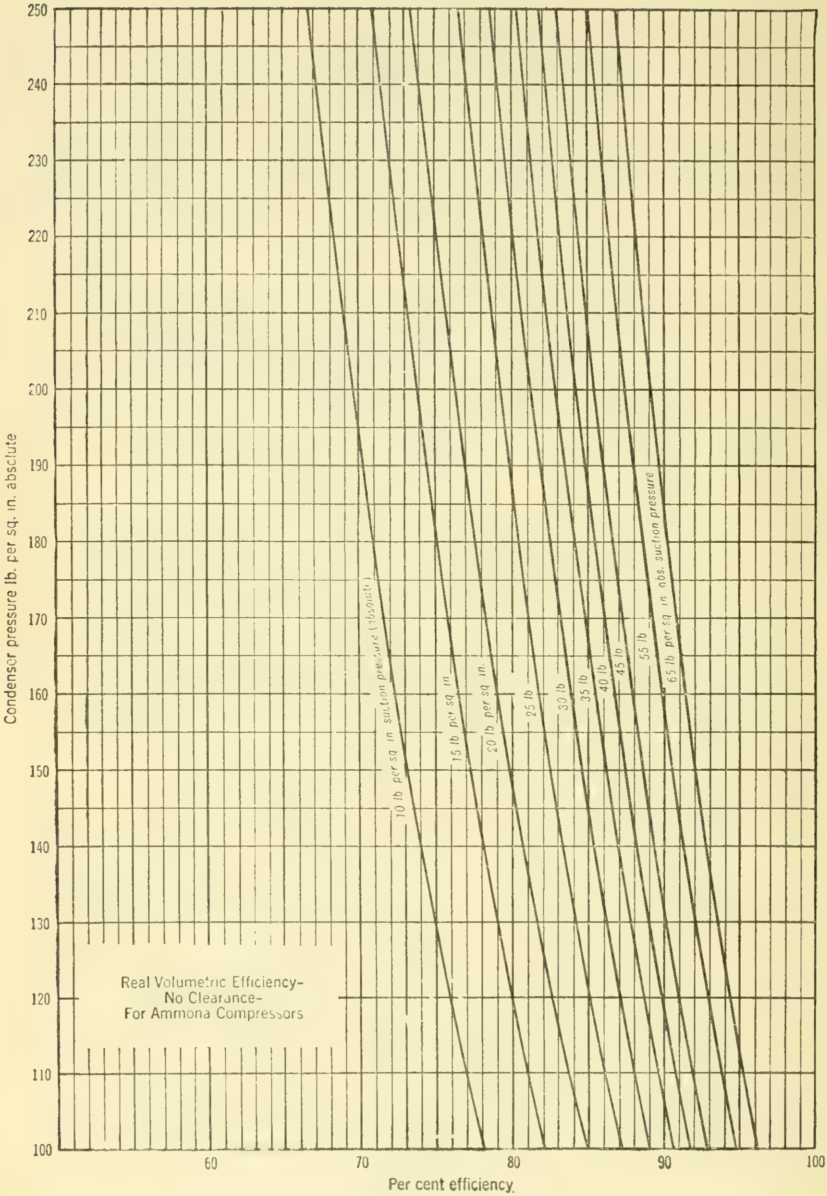


FIG. 22.—Volumetric Efficiency of Ammonia Compressors.

The Water Jacket.—The effect of the water jacket for ammonia compressors has been given considerable attention, although until 1921 it was a matter of controversy between manufacturers and theorists.

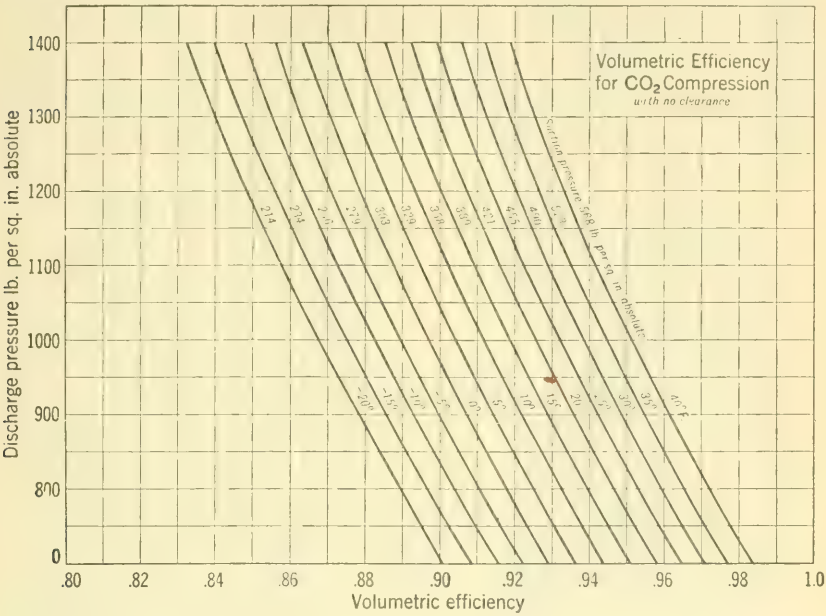


FIG. 23.—Volumetric Efficiency of Carbonic Compressors.

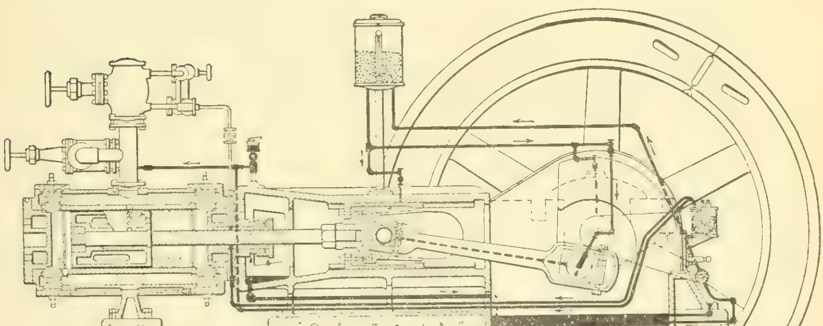


FIG. 24.—Lubrication; the Horizontal Compressor.

The accepted opinion has been that the water jacket is of little value as regards its effect on the compression line, but that its advantage is mainly in reducing the discharge gas temperature and in preventing the overheating of the compressor cylinder.

Because of the fact that the suction gas enters at a temperature much lower than that of the jacket water, the water jacket is designed slightly different from that in the air compressor. The vertical single-acting ammonia compressor is water jacketed the last third of the stroke, whereas the horizontal double-acting compressor is jacketed frequently only at the heads and the ports. The carbonic compressor usually has not been jacketed at all because of the comparatively small amount of rise of temperature; and the majority of the fractional tonnage compressors, using sulphur dioxide or methyl chloride, are designed for air cooling.

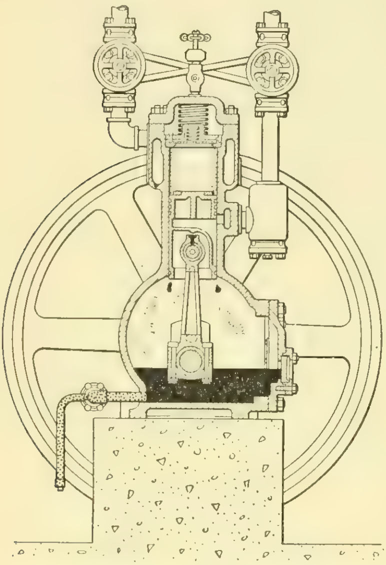


FIG. 25.—Lubrication; the Vertical Enclosed Compressor of 25 Tons and Less.

The most complete tests on the effect of water jackets are those of Walter Fisher.[3] These tests show the relative effect on the compressor with and without the water jacket. For example it is shown that the real volumetric efficiency is increased from about 1.5 to 4.0 per cent by having the water jacket.

Lubrication.—With the changes in design have come radical changes in the manner of lubrication. With the horizontal and the semi-enclosed vertical compressor the design has developed into an automatic forced feed system for each surface and bearing, embodying also the oil filter system (except for cylinder lubrication) in order to keep the oil in good condition. As the speed of the compressor increases, the need of a positive, correct method of lubrication is very evident. In the case of the crosshead pin the lubrication is frequently obtained by running the oil pipe along the connecting rod (Fig. 24) in some suitable manner from the crankshaft, or by drilling along the entire length of the rod as shown in Fig. 18.

The enclosed type of single-acting vertical compressor has been lubricated only by the splash method (Fig. 25) and this means is still used in the smaller compressors. In the splash system of lubrication the oil in the crankcase is splashed at each revolution of the crank.

[3] Walther Fisher—Der Einfluss des Kuhlwassermantels an Kompressions-Kältemaschinen—Verein deutsche Ingeniere, 1921.

The splash method of lubrication is excellent for certain conditions, and it is the cheapest of all methods used for continuous work. There are two objections to its use. The first objection is that at the higher speeds the lubrication may not be positive enough because cavitation results in an inefficient formation of the splash with the result that lubrication may be poor or even a failure. The second objection is that lubrication depends on the continuous presence of oil in the crankcase. Certain operating conditions assist in the loss of this oil through

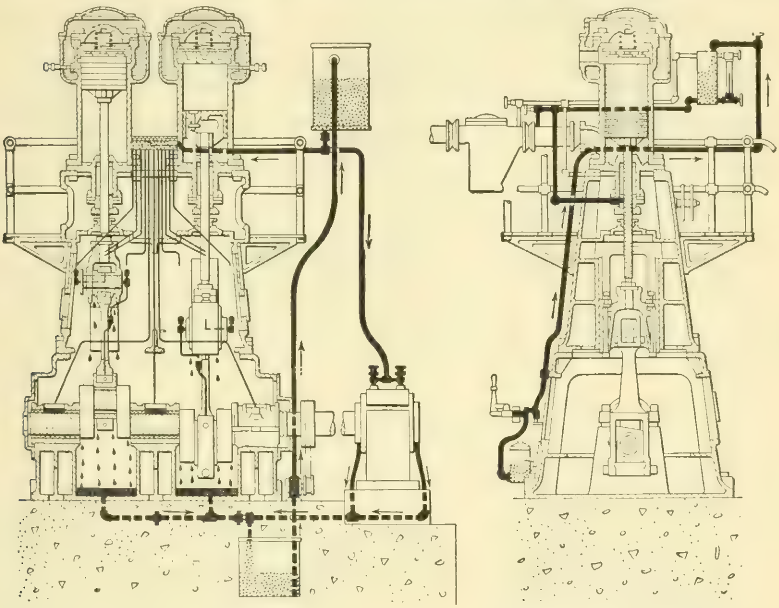


Fig. 26.—Lubrication; the Vertical, Open Frame, Compressor.

the piston into the oil separator, leaving the crankcase with insufficient oil. There is also the possibility of pumping oil during normal operation, but this danger is lessened by care in the design of the rings in the piston.

The result of this lack of positive lubrication has resulted in the design of the forced feed type of pump lubricator, use being made of the oil filter and (usually) a spy glass in the pipe line to the several bearings. Figure 15 shows this type of lubrication design as applied to the smaller capacity of compressor, say from 2 to 12 tons of refrigeration and designed for speeds varying from 530 r.p.m. at 2 tons to 400 r.p.m. at 12 tons. This design lacks the filter and the sight feed and is entirely within the crankcase, but the action may be made positive,

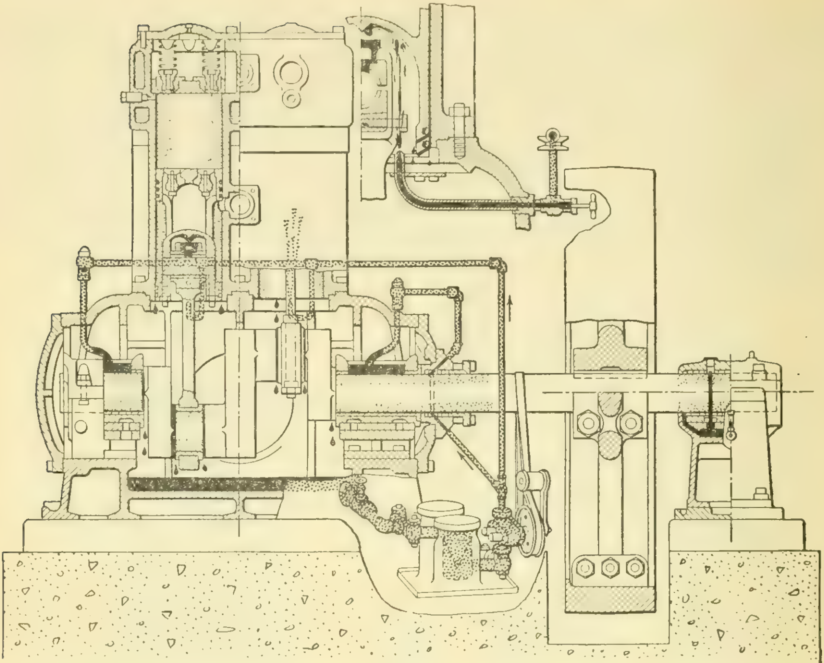


FIG. 27.—Lubrication; the Large, Enclosed-type, Ammonia Compressor.

except for clogging, by the use of some form of rotary pump and by having no suction lift to the pump itself. Figure 27 shows a design typical of the more elaborate method of lubrication, such as would be found in the larger compressors, one which has a nozzle device for spraying the cylinder, provided with a nozzle clearer, to be used at regular intervals to prevent the clogging of the nozzle, and a separate pipe line for the oil to every important bearing. Figure 26 shows the method used in the vertical open-frame compressor.

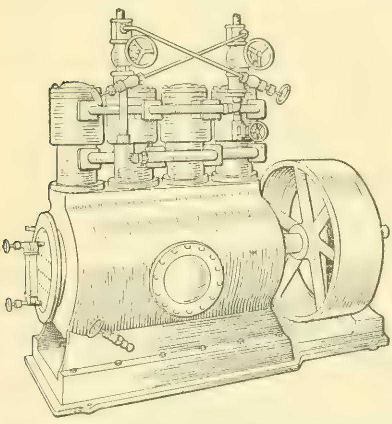


FIG. 28.—The Eccentric Driven Compressor.

Number of Cylinders.—The use of multiple cylinders has been restricted in the United States to two in number as a rule, although the

three-cylinder compressor has been built to a limited extent. This is apparently a contradiction of the practice in Europe and particularly in Great Britain, where they are pushing the type using the three- and the four-cylinder design of compressor especially in the larger sizes, as shown by Fig. 18. In Great Britain the tendency appears to be for increase of the capacity of the compressor by adding cylinders, but in the enclosed design only and not the open frame semi-enclosed type of single-acting vertical compressor—the latter being distinctively an American design. There is another advantage in the use of multiple cylinders from the viewpoint of balance. With one or two cylinders no amount of counterbalance will entirely balance the effect of reciprocation whereas the three- or four-cylinder machine (Figs. 18, 19, 28) may be entirely balanced except for a moment which will need to be carried by the bed plate or the foundation. It would seem then that if the compressor speed is to be increased decidedly, there will be an advantage in the use of three cylinders on the same crankshaft.

Theoretical Displacement of the Compressor.—The theoretical displacement of the compressor may be found in the following manner:

The refrigerating effect of the volatile liquid per 1.0 lb. of refrigerant is $(i''_3 - i'_1)$ (Fig. 1). The piston displacement per ton of refrigeration per minute

$$\text{P.D.} = \frac{200}{i''_3 - i'_1} \times V''_3,$$

where V_3'' is the specific volume of saturated ammonia (or refrigerant) at the pressure *3*. If the real volumetric efficiency of the compressor (expressed as a decimal) is E_s, then the total required piston displacement becomes

$$\text{P.D.} = \frac{200}{i''_3 - i'_1} \times V''_3 \div E_s$$

assuming tight valves, rings and no clearance. If E_v is the volumetric efficiency due to *clearance only*, then the piston displacement required with clearance is,

$$\text{P.D.} = \frac{200}{i''_3 - i'_1} \times \frac{V''_3}{E_v \times E_s}$$

There is no reliable method of calculating the real volumetric efficiency although the formula $E_s = 1 - \frac{t_4 - t_3}{1330}$ where $(t_4 - t_3)$ is the rise of temperature during the compression stroke has been proposed

as an empirical formula based on tests conducted by the York Manufacturing Co. (1903) and gives approximate results. The temperature at the point *4* can be found from the thermodynamic equation for exponential processes (Chapter I):

$$T_b = T_a\left(\frac{p_b}{p_a}\right)^{\frac{n-1}{n}}$$

where the temperatures are absolute and $t_b = T_b - 460$.

Figures 29 and 30 have been calculated, using the equations for volumetric efficiency and piston displacement.

Horse Power per Ton of Refrigeration.—The horse power per ton of refrigeration is a very important quantity, and the theoretical value may be found in the following manner:

The work during any exponential compression cycle is:

$$W = \frac{n}{n-1}p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

where n is taken as approximately 1.28 for ammonia, and for carbon dioxide, and p_1 and p_2 are the absolute pressures at the beginning and the end of compression in pounds per square foot and v_1 is the volume in cubic feet at the beginning of compression. If v_1 is taken in the formula as the volume of the piston displacement per ton of refrigeration per minute, and the result is divided by 33,000, the final result is the horse power per ton of refrigeration. For ammonia and carbon dioxide, and several other refrigerants, there is also the P–I diagram (Fig. 164, etc.) from which it is possible to get the work of compression per one pound of the refrigerant, by reading the value of the thermal potential I at the two points corresponding to the beginning and the end of compression. The theoretical indicated horse power (i.hp.) can then be found by multiplying by a suitable constant and by the number of pounds of refrigerant required per ton of refrigeration per minute. The formula becomes

$$\text{i.hp.} = \frac{200}{i''_3 - i'_1} \times \frac{777.6}{33{,}000} \times (i_4 - i_3) = 4.713\left(\frac{i_4 - i_3}{i''_3 - i'_1}\right)$$

where $i_3'' - i_1'$ is the net refrigeration per 1.0 lb. of refrigerant.

Figures 31 and 32 give values for the horse power per ton of refrigeration for ammonia and carbon dioxide compressors.

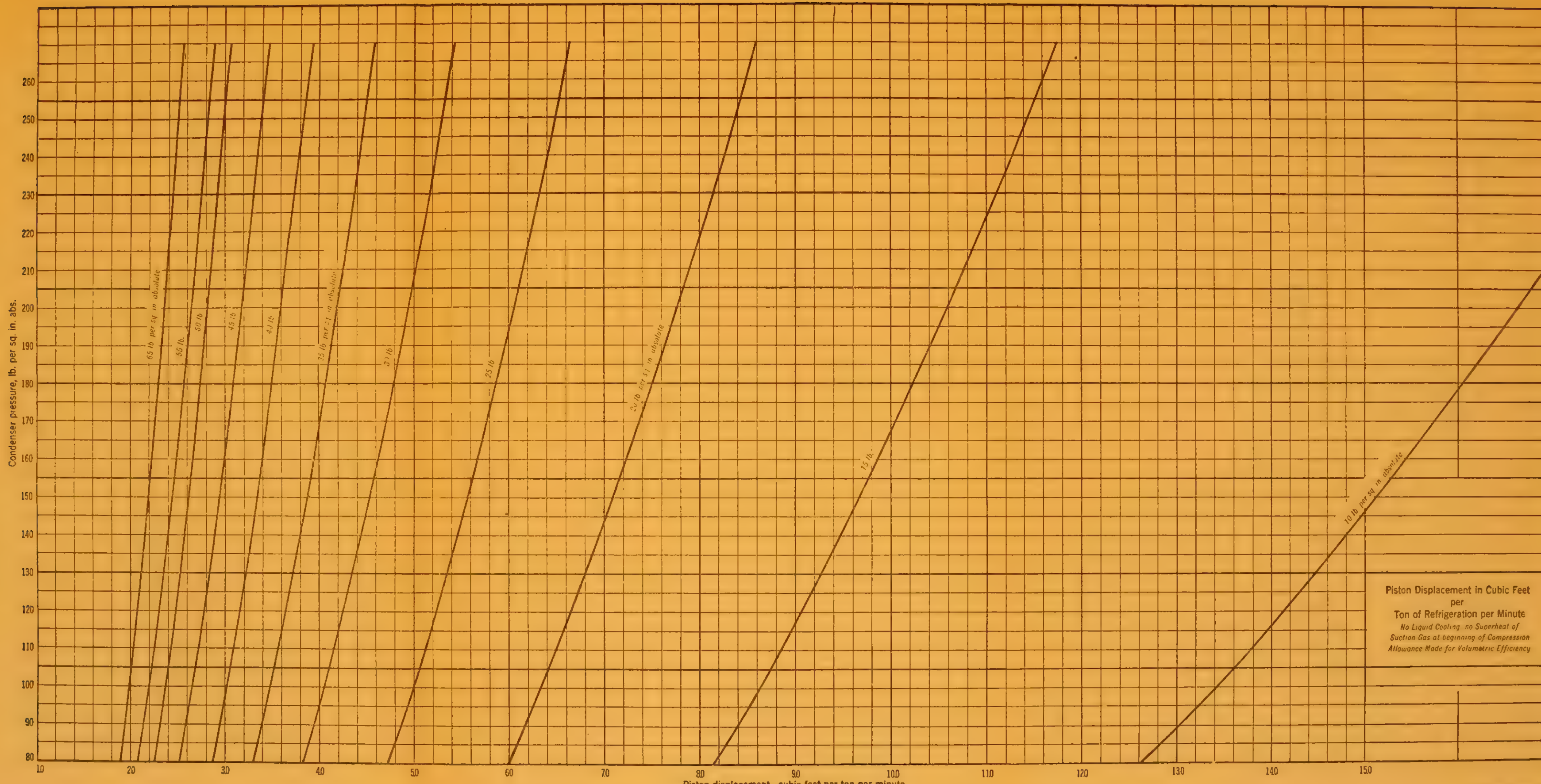


FIG. 29.—Piston Displacement of Ammonia Compressors, per Ton of Refrigeration per Minute.

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CARBON DIOXIDE COMPRESSORS

The use of carbonic refrigeration is increasing in the United States but not in proportion to the amount of new tonnage sold. Contrary to the conditions in Great Britain, where carbonic refrigeration is required by law under specific conditions, carbonic refrigeration is making gains in the United States practically on its own merits. It is a real *safety* refrigerant, and as such is particularly advantageous for use in theaters, hotels, apartment houses and other places where the

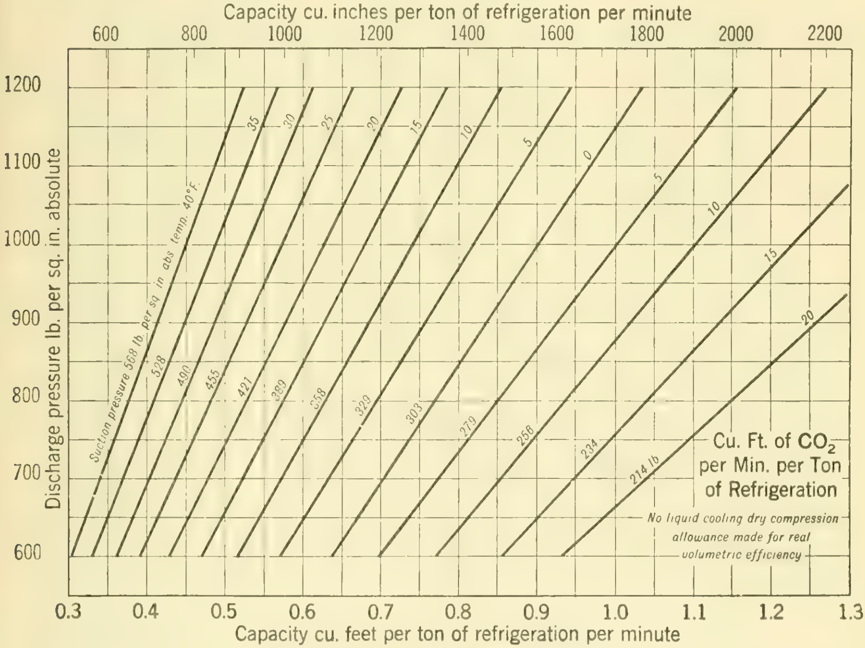

FIG. 30.—Piston Displacement of Carbonic Compressors, per Ton of Refrigeration per Minute.

public is exposed, and the risk due to panics might be a positive danger. A small amount (even a trace) of ammonia in the air is a menace, whereas it has been demonstrated that carbon dioxide in large quantities is not injurious except in that it excludes oxygen, a condition that might finally cause suffocation.

Carbonic refrigeration has suffered because of an improper understanding of the refrigerant. As the critical temperature is 88.0 deg. F. (corresponding to a pressure of 1072 lb. per square inch) it is evident that temperatures of the condensing water of 80 deg. F. and over, mean

that the refrigerant is *not* condensed, but that does not mean that no refrigeration is possible under these conditions. It means only that less useful refrigeration is possible per pound in the same manner that less refrigeration is possible when a hot liquid passes the expansion valve in ammonia compressors than would be the case with a cold liquid. It seldom occurs that the quality (x) of the gas after the expansion valve even under these extreme conditions is less than 0.5, as is seen on referring to the *P–I* diagram for CO_2 (Fig. 164).

There is another factor antagonistic to the increased use of carbonic machines. The pressures involved are heavy, varying from 1000 to 1200 lb. or more on the high-pressure side and from 300 to 600 lb. per square inch on the low-pressure side. Losses of gas to the atmosphere have been excessive, and leaks past the valves, piston rings, etc., have been very high even with new machines. It appears then that a successful carbon dioxide compressor and system must be properly designed, and the materials entering into its construction and the *workmanship* must be satisfactory. Fortunately autogenous welding has become so common and reliable, especially for small pipe work, that fittings can be eliminated to a large extent, and the gas losses can be reduced accordingly. This is becoming true in the United States for the double-pipe condenser (Fig. 66), the low-pressure piping and to some extent the high-pressure part of the system.

Design.—Carbonic compressor cylinders are made of semi-steel (instead of steel forgings as formerly) cast solid and bored out for the cylinder bore, the ports and valve cages. As a rule the cylinder, crank-end cylinder head and the stuffing box are made of a single casting, thereby permitting better alignment, and eliminating one joint and the source of another probable leak. The head cylinder cover is held in place by studs or through bolts. The remainder of the construction is very much like the design for ammonia compressors, as regards construction of the piston rod, connecting rod, crosshead, etc. For example, the piston rods, connecting rods and crankshaft are of forged steel, the piston rod being hardened and ground to size. The crosshead is of semi-steel and the pin is of steel hardened and ground. The brasses are of phosphor-bronze and are made adjustable. Because of the small ratio of the condenser to the suction pressure $\left(\frac{p_2}{p_1}\right)$ the discharge gas is not high in temperature and water jackets for the cylinder are never used, but on account of the very heavy discharge pressure, and the desire to keep the so-called hoop tension in the cylinder as low as practical—the cylinder diameter is kept as small as possible. The necessary piston displacement of the compressor is obtained by making the stroke

long in proportion to the piston diameter, the ratio being from $3\frac{1}{2}$ to 4, or more than twice the ratio usually found in similar ammonia compressors. Except for marine and some small enclosed machines the compressor is of the horizontal, double-acting type, using metallic packing and the same design of lantern as in ammonia compressors. Also, because of the small ratio of pressures, clearance does not affect the capacity as much as it would in the case of ammonia, and the capacity is not reduced very much by wire drawing in the suction because of the usual high suction pressures of 300 to 450 lb. In consequence a drop of 5 lb. due to friction would not be as serious as would an equal drop in the case of ammonia with a suction pressure of 15 lb. gage. The valves may be poppet valves in 45-degree seats, or they may be ring plate valves, in which case, the disc is frequently made of chrome vanadium steel, and the lift is limited to as little as $\frac{1}{16}$ in.

Although the ratio of the condenser to the suction pressure is not great, being but seldom more than 1 : 4, yet the differential pressure is very heavy—being from 900 to 1000 lb. The cylinder and the piston should be ground carefully to a true surface and the cylinder should be lapped out. The piston is supplied with cast iron or semi-steel snap rings which may be 3, 4, 5, or 6 in number. The H. J. West design of vertical enclosed type of compressor which is designed for rotative speeds of 400 to 500 r.p.m. has 7 snap rings in addition to 2 scrapper rings placed on the crosshead part of the trunk piston. The tolerance for cylinder bore is 0.002 in. for diameter and parallel to 0.001 in. per foot of length. The piston diameter is specified to be at least 0.0005 in. per inch of diameter less than the nominal diameter of the piston, but in no case is the difference between the cylinder diameter and the diameter of the piston to be less than 0.0007 in. per inch of diameter, nor more than 0.0015 in. per inch of diameter. It is made round and parallel within the limits of 0.001 in. The piston rod is round within the limits of 0.002 in. and parallel within the limits of 0.003 in. The total clearance between the piston head and the ends of the cylinder is about $\frac{1}{8}$ in. The entire system is designed to be tight under a pressure of 3000 lb. per square inch of water and 1350 lb. air pressure under water.

The stuffiing box is still designed to use leather cups and other soft packing combined with brass boxes and washers or other forms of distance pieces, but this design is giving way to metallic packing. When the soft packing is used, oil is fed into the lantern at a pressure greater than the gas pressure in the gland and the stuffing box is well supplied with oil so that the surplus works along the rod and into the cylinder, lubricating the piston in the cylinder. Metallic packing is usually freer from leaks provided the rod is turned true and is centrally located

in the machine. There is less danger of burning the packing than in the case of the leather cups.

The carbonic compressor has two factors influencing its capacity, one of which, the leakage factor, has been discussed already. Leaks past the valves and the piston are reduced to a minimum by the use of good material and design and very careful machine work. The second factor is the large value of the thermal potential (i) of the liquid and the relatively low value of the latent heat of vaporization. Referring again to the *P–I* diagram (Fig. 164), it will be seen that the value of "x" is quite large even for a nominal range between the suction and the condenser pressures. The result is that whereas the power requirement would be the same for all refrigerants if the liquid is cooled down first to the temperature of the evaporator, yet, as a matter of fact, there are large variations. Tests reported by Professor Carl Linde[4] showed, with 72 deg. F. condenser temperature and 15 degrees brine, that the carbonic machine was only equal to 82 per cent of the coefficient of performance of an ammonia machine operating under similar conditions, and if the condenser temperature was increased to 95 degrees the coefficient of performance dropped off to 50 per cent. Willcox and Hodgdon showed[5] that the brake horse power per ton of refrigeration (i.hp. plus ten per cent) with − 10 deg. F. evaporating temperature of the liquid carbon dioxide was 1.67 with 55 deg. F. liquid CO_2 at the expansion valve, 2.0 with 65 degrees, 2.39 with 75 degrees, 2.85 with 85 degrees and 3.44 b.hp. with 95 degree "liquid" at the expansion valve per ton of refrigeration. These values are considerably higher than would be required with ammonia. The natural consequence is that ammonia is used to the exclusion of carbon dioxide in all large cold-storage warehouses or in ice plants where the unit power costs—the cost of power per ton of refrigeration per 24 hours—or the power required per ton of ice delivered on the platform—is an important factor. On the other hand, some hotels make their own ice, with the refrigerating plant installed for cooling drinking water, and for kitchen and storage boxes. Therefore carbonic compressors are frequently used for this kind of installation, even though the cost of making the ice is greater than it would be with ammonia as the refrigerant. Figs. 23, 30 and 32 give an idea of the volumetric efficiency, the cubic feet of piston displacement per minute per ton, and the brake horse power per ton of refrigeration. Fear of ammonia is so great with certain individuals that carbonic compressors are becoming more popular in America and without doubt improved design will make them even more so. Installa-

[4] Zeitschrift für die Gesammte Kälte-Industrie, Jan. 28, 1895.

[5] Willcox and Hodgdon, Cold Storage and Ice Association, Jan. 27, 1914.

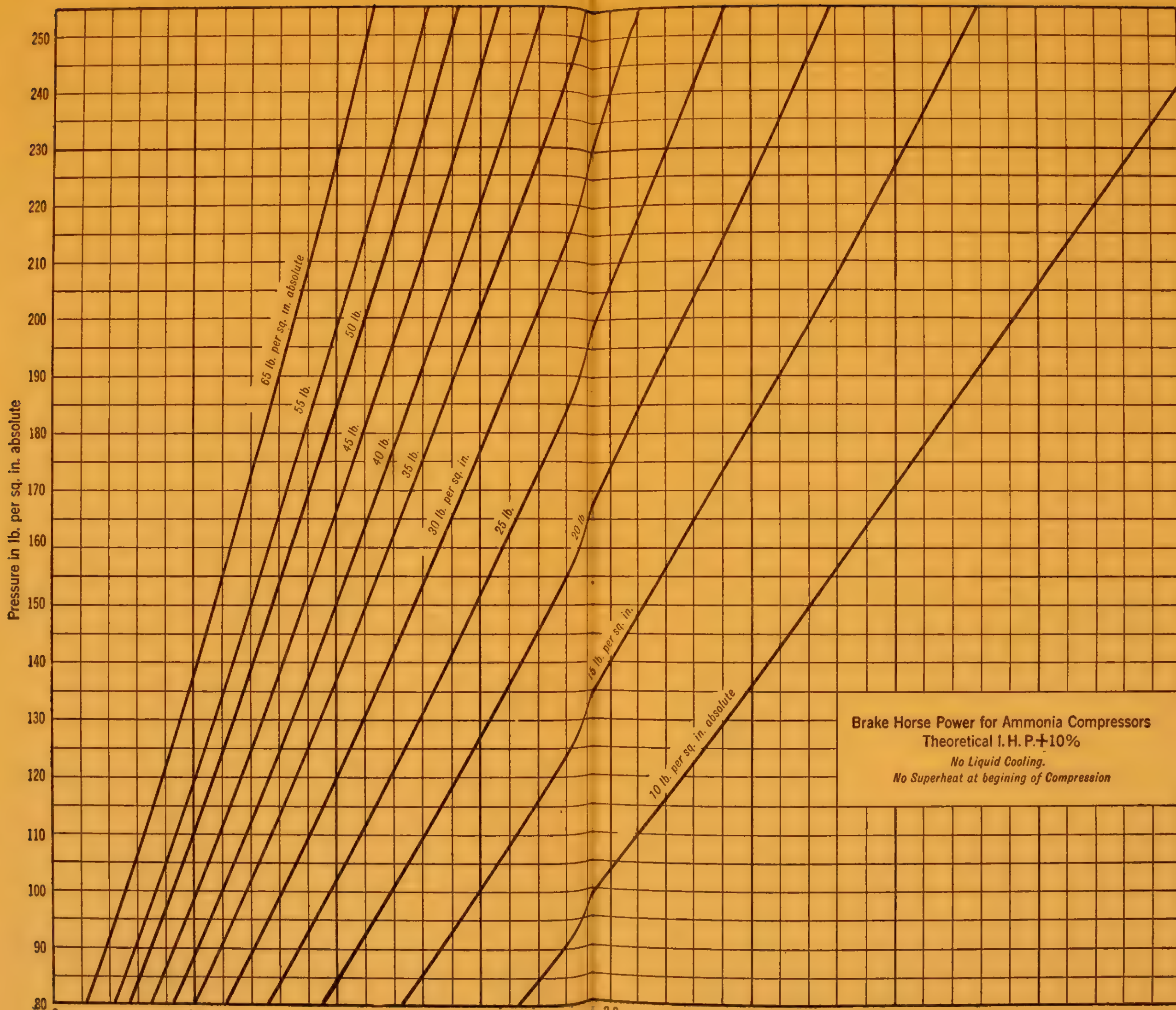

FIG. 31.—Horsepower per Ton of Refrigeration for Ammonia Compressors.

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tions are being made even in Florida and the Gulf States, though the cooling water necessary for this type of machine is of a temperature that would seem prohibitive in such latitudes.

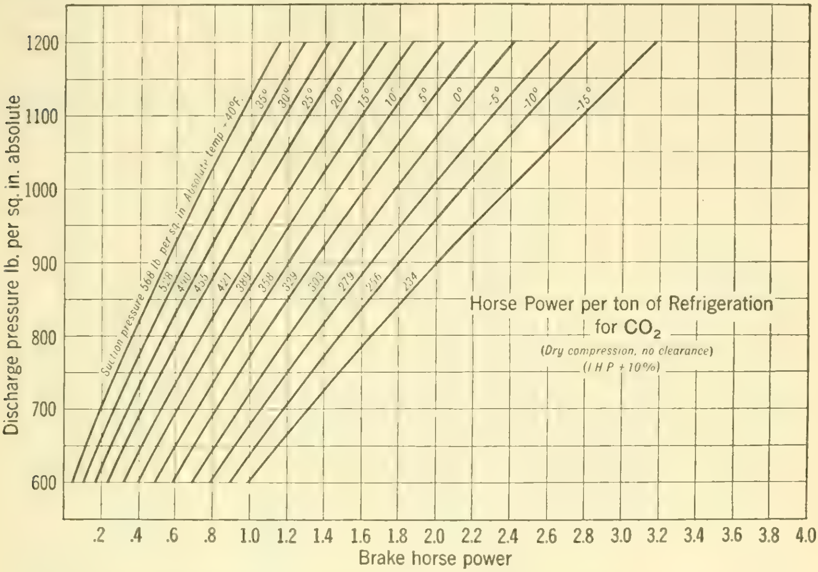


FIG. 32.—Horse power per Ton of Refrigeration for Carbonic Compressors.

The carbonic compressor has been used for many years in the merchant marine to the exclusion of all others, on account of the Underwriters' rules. Lately, however, there has been a tendency for the use

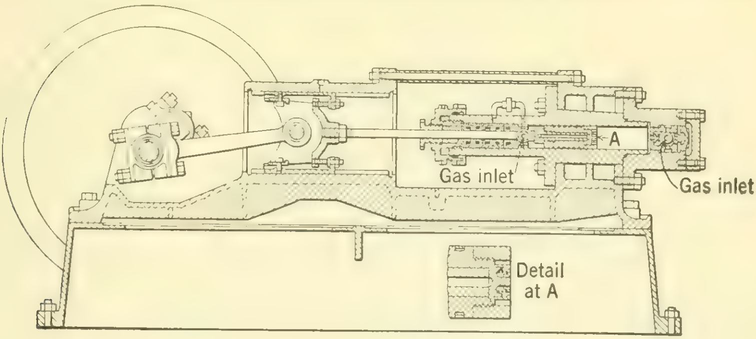


FIG. 33.—The Carbonic Compressor.

of ammonia compressors located outside the main engine room and preferably convenient to the deck, especially if the size of the installation warrants the employment of a refrigerating engineer. In the smaller

sizes, where the regular watch engineer has to operate the refrigerating machine also, the installation has to be in the main engine room. Where ammonia compressors are used on shipboard wrought iron and steel condensers have to be selected, in which case corrosion is more rapid than would be the case of copper condensers which are employed so extensively for carbonic installations.

The carbon dioxide compressor can produce low temperatures with considerable ease and in Great Britain they have been built in two stages for that purpose. According to G. W. Daniels,[6] a battery of 4

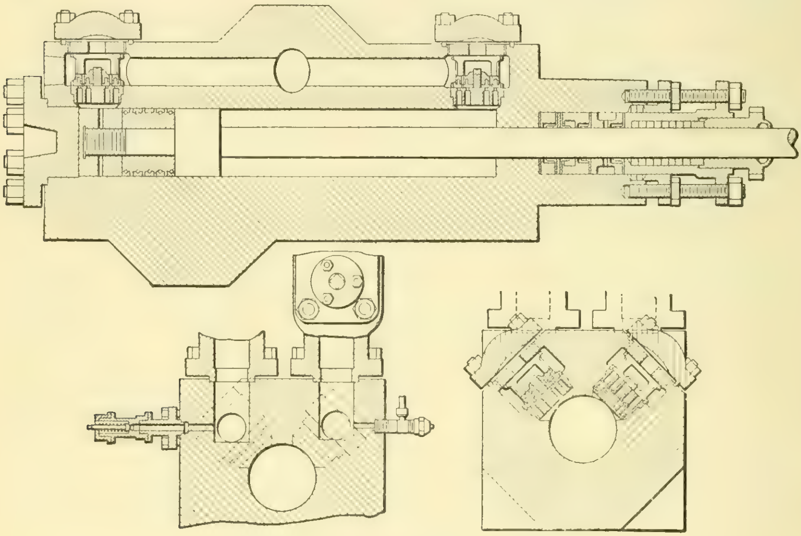


Fig. 34.—The Carbonic Compressor.

2-stage carbonic compressors operating at − 58 deg. F. has been very successful. Figures 33 to 35 inclusive show representative carbonic compressors and Fig. 36 gives results of tests on a small marine type carbonic compressor.

The Sulphur Dioxide Compressor.—The compressor using sulphur dioxide as a refrigerant has not been used to any extent in the United States except in the case of household machines, although numerous attempts have been made to establish its use. This fact is somewhat surprising when one recollects that in Germany alone there are 17 firms listed as building SO_2 machines, including A. Borsig, Quire and Co., Gesellschaft für Lindes Eismaschinen, Maschinenbau-Anstealt Humbolt, etc., built in capacities up to 320,000 cal. per hour or about 106 tons of

[6] G. W. Daniels, Refrigeration in the Chemical Industry. A. J. Rayment, 1925.

refrigeration, as compared with 27 German firms building ammonia and 29 building carbonic compressors. In a letter in the May, 1925, issue of Cold Storage, W. S. Douglas of Wm. Douglas & Sons says that 90 per cent of all SO_2 refrigerators operate at a brine temperature of 15 to 25 deg. F. He says the use of SO_2 as a refrigerant is increasing steadily and, in his opinion, this is the best refrigerant for small machines on account of the low pressure, the lubricating properties and the ease in handling.

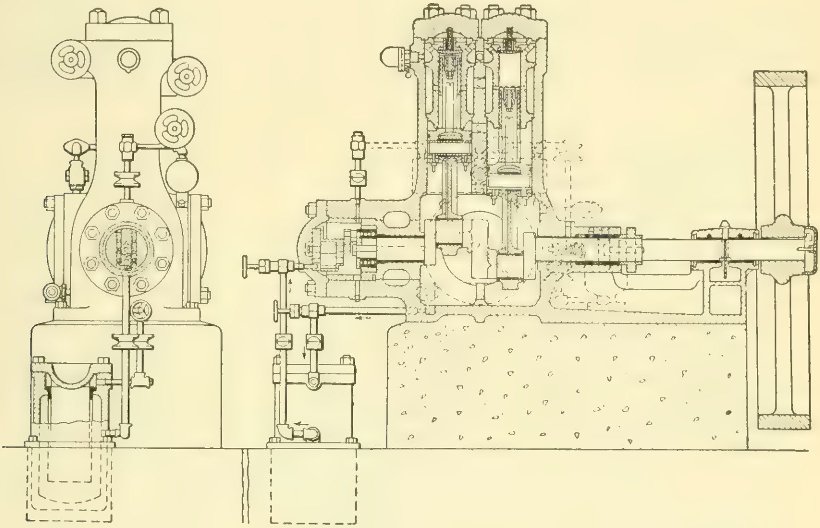


FIG. 35.—The Enclosed Type Carbonic Compressor.

The sulphur dioxide compressor requires some 2.5 times the piston displacement of the ammonia compressor, and the design of the German compressors very closely resembles that for ammonia. Because of the larger volume of the gas handled, and the relatively low unit pressures the valve is likely to be of the flap (Gutermuth) type as used in the larger machines of the A. Borsig design or of the modified plate-valve type. As sulphur dioxide is corrosive when any moisture is present the tendency is to restrict the amount of evaporator surface as much as possible, and in consequence the direct expansion system is not popular but a self-contained brine system is more likely to be used. In the United States it seems now very unlikely that sulphur dioxide will be used to any extent except in small machines of the household or the $\frac{1}{6}$-ton or the so-called ice cream cabinet size or in the case of the *A-S* (dumb-bell) (Fig. 37 and Table 2) compressor. In this compressor the casing is

hermetically sealed and cannot be opened for repairs except at the factory. Also, except in the small sized compressor, sulphur dioxide has practically nothing in its favor, and it is more than likely that it will be abandoned in favor of methyl chloride, butane or some other low-pressure refrigerant. All of these have the advantages, except for self-lubrication, and none of the disadvantages of sulphur dioxide, although it is true that sulphur dioxide (SO_2) leaks, being easily detected, can be easily remedied.

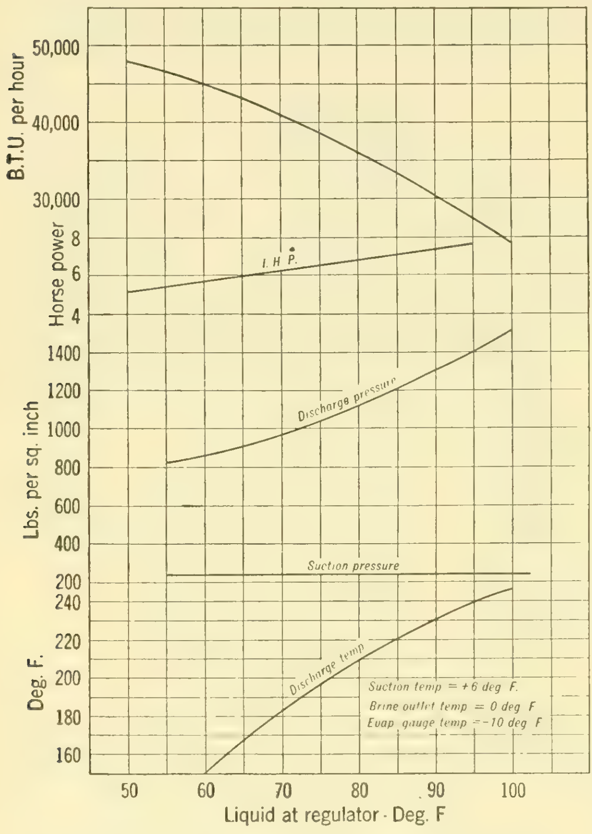


FIG. 36.—Test Performance of Carbonic Compressors.

The Rotary Compressor. — The rotary compressor is the ultimate goal of many engineers as it will permit high rotative speeds and therefore direct connection to a cheap type of electric motor. The piston displacement per ton of refrigeration need not cause any concern as the displacement is easily cared for in such a design, and the refrigerant may be chosen that is best suited to rotary compression without regard to other factors.

The Blade Type.—The most common type of rotary is the blade type to operate eccentric in the casing and so arranged as to depend on centrifugal force to keep the blade always in contact with the casing. Such a machine using ammonia has been described by Roloff,[7] but no such compressor has been successful. Other compressors (Fig. 38), using ethyl chloride, have been successful on test. Results of these

[7] Roloff, Transactions of the A. S. M. E., 1918.

tests reported by Herter[8] indicate that the economy of operation is not so great as for ammonia, although it must be remembered that the small machine is seldom as efficient as the larger one. His figures are as follows:

Refrigerant evaporation temperature, degrees F.	30	19.9	2.7	−9.6
20-ton ammonia compressor (horse power per ton)	1.1	1.3	1.6	2.0
2-ton Clothel compressor (horse power per ton)	2.3	2.8	3.3	4.4
$\frac{1}{10}$-ton Williams compressor (horse power per ton)	4.3	5.8	11.0	22.0

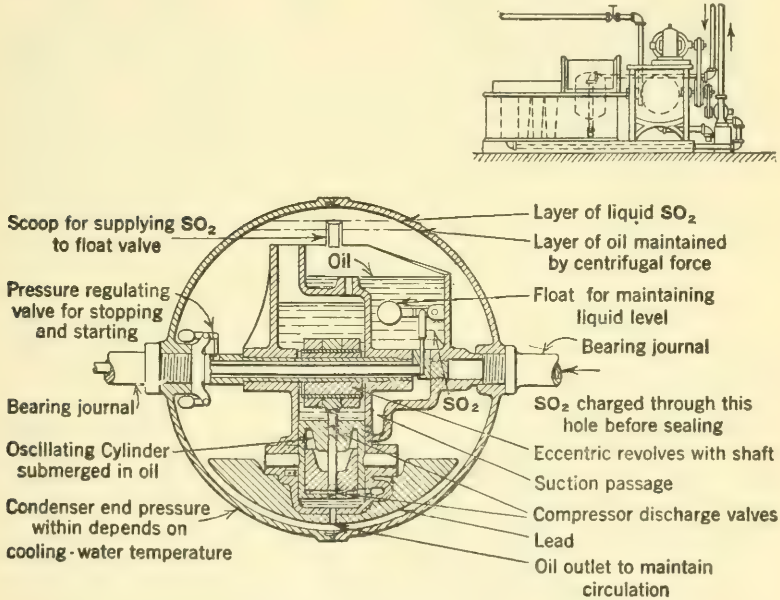


FIG. 37.—Audiffern-Singrun Sulphur Dioxide Compressor.

The Centrifugal Compressor.—If the centrifugal compressor is used, the design principle is that of developing a velocity to the gas, and then by slowing up this velocity of converting velocity head into pressure head. Such a method has been used to compress air by the use of a very large number of stages. For refrigeration the compression of carbon dioxide and ammonia is practically impossible, but by the judicious selection of a heavy gas of low condensing pressure the same end can be reached. This has been accomplished by the Carrier

[8] C. H. Herter, Refrigerating World, Sept., 1922.

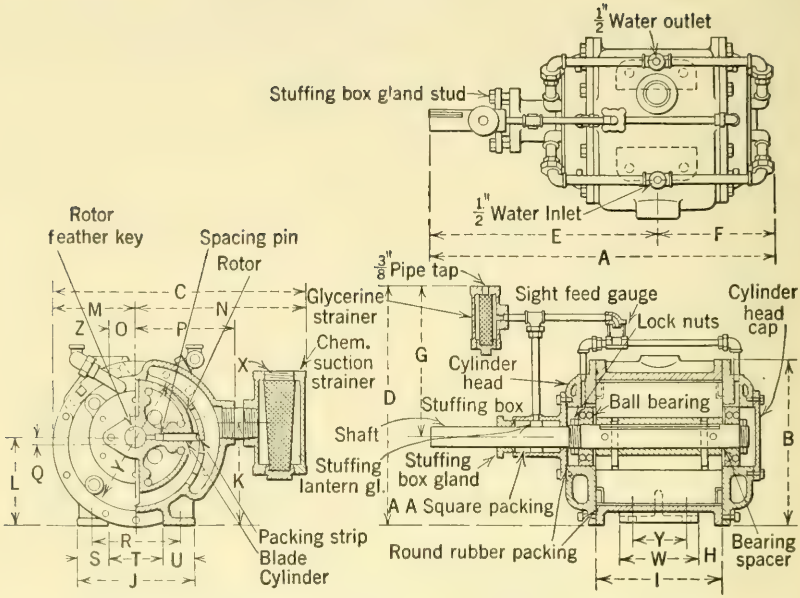


FIG. 38.—The Clothel Rotary Compressor.

Size	Dimensions in Inches							
Tons	A	B	C	D	E	F	G	H
$\frac{1}{2}$ 1 2	$15\frac{1}{2}$ $25\frac{1}{8}$ $32\frac{3}{4}$	$6\frac{5}{8}$ $10\frac{5}{8}$ 16	$12\frac{5}{16}$ $17\frac{5}{16}$ $24\frac{3}{8}$	$14\frac{3}{4}$ 18 23	$10\frac{9}{16}$ $16\frac{1}{4}$ $21\frac{5}{8}$	$4\frac{15}{16}$ $8\frac{7}{8}$ $11\frac{1}{8}$	11 12 $14\frac{1}{2}$	$\frac{13}{32}$ $\frac{9}{16}$ $\frac{11}{16}$
Tons	I	J	K	L	M	N	O	P
$\frac{1}{2}$ 1 2	$3\frac{1}{2}$ 9 12	6 10 $11\frac{1}{2}$	$3\frac{13}{32}$ $6\frac{7}{8}$ $9\frac{7}{8}$	$3\frac{3}{4}$ 6 $8\frac{1}{2}$	$3\frac{5}{16}$ $5\frac{5}{16}$ 8	9 12 $16\frac{3}{8}$	$1\frac{1}{2}$ $1\frac{3}{4}$ $2\frac{1}{2}$	4 $6\frac{1}{4}$ $9\frac{1}{2}$
Tons	Q	R	S	T	U	V	W	X
$\frac{1}{2}$ 1 2	$\frac{7}{32}$ $\frac{7}{16}$ $\frac{21}{32}$	$4\frac{3}{4}$ 8 $9\frac{1}{4}$	$1\frac{1}{2}$ $2\frac{1}{4}$ 3	3 $5\frac{1}{2}$ $5\frac{1}{2}$	$1\frac{1}{2}$ $2\frac{1}{4}$ 3	$1\frac{1}{4}$ $4\frac{1}{4}$ $5\frac{3}{4}$	$2\frac{1}{2}$ $5\frac{3}{4}$ $7\frac{1}{2}$	1 2 $2\frac{1}{2}$
Tons	Y	Z	A–A	R. P. M.	Net Weight, Pounds			
$\frac{1}{2}$ 1 2	5 8 $12\frac{1}{4}$	$\frac{3}{4}$ $1\frac{1}{4}$ 2	$\frac{5}{16}$ $\frac{3}{8}$ $\frac{1}{2}$	900 350 226	65 lbs. 225 lbs. 580 lbs.			

Engineering Company using as a refrigerant dichlorethylene ($C_2H_2Cl_2$) which condenses at a temperature of 80 deg. F. at a pressure of less than one atmosphere. The compressor (Fig. 39) looks very much like

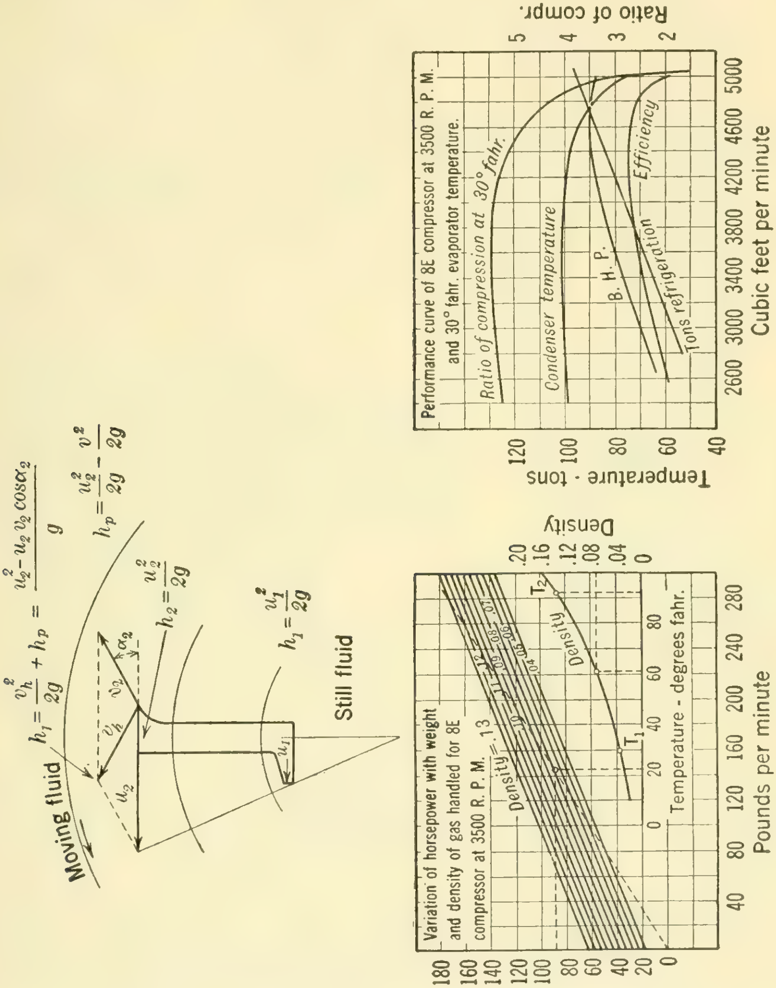


Fig. 39.

an ordinary water centrifugal pump and is designed for five, six or more stages and about 3600 to 4000 r.p.m. in some sizes. This type of compressor appears to be best suited to the larger sizes, say 75 tons and larger, but it may be possible to build economically in capacities as low as 20 tons. Also the machine is best suited to the higher evaporating

temperatures such as would be found in the cooling of air for theaters and public buildings.[9]

Referring to Fig. 39 it will be seen that the total head produced, the sum of the velocity and static heads, is

$$h_1 = \frac{v_h{}^2}{2g} + h_p = \frac{u_2{}^2 - u_2v_2 \cos a_2}{g}$$

Only a part of the velocity head is available for the production of static pressure, and the usual method of change from velocity to static heads is by means of diffuser blades of gradually increasing cross-sectional area. The efficiency of the conversion is approximately 70 to 80 per cent.

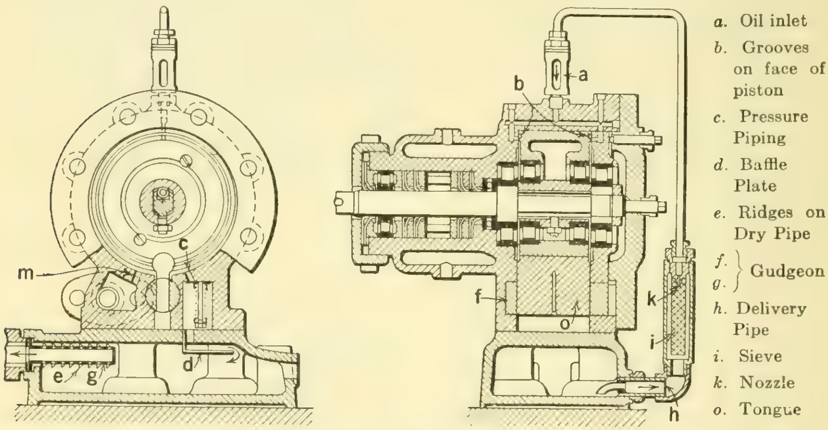


FIG. 40.—The Güttner Pendulum Compressor.

The curves in Fig. 39—taken from an article by Willis Carrier (Refrigerating Engineering, Feb., 1926)—give some idea of the displacement, tonnage and efficiency of a centrifugal compressor using " dieline " as a refrigerant.

Turbo-Compressors.—In Germany[9a] the turbo-compressor seems to be successful in the larger sizes. According to Heinz Voight ammonia turbo-compressors are only applicable in capacities greater than 300 tons. The number of rotors for different refrigerants with peripheral speeds of 655 ft. per second are stated to be as follows:

[9] Because the evaporator pressure is so low, a separate means of continually expelling the air leakage must be provided. The float type of expansion valve is used. The compressor, condenser and brine cooler are made self-contained.

[9a] Heinz Voight, Zeitschrift des Vereins Deutscher Ingeniere, Aug. 13, 1927.

Number of Rotors Required for Different Mediums at Peripheral Speeds of 655 Ft. per Second

Temperature of Vaporization, Deg. F.	CO_2	NH_3	CH_3Cl	SO_2	C_4H_{10}	C_2H_5Cl	H_2O
32	2	6	3	3	3	3	17
14	3	11	4	4	4	4	
− 4	4	15	5	5	5	5	
−22	5	20	6	6	6	6	

The guaranteed power consumption of a 2000-ton turbo-compressor for operation at 5 deg. F. evaporation and 86 deg. F. liquefaction temperature at a speed of 6000 r.p.m. is given as 2365 b.hp.

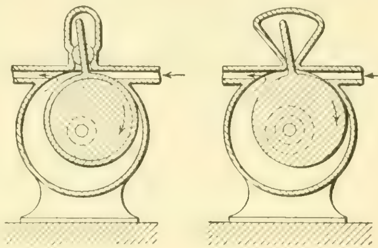


Fig. 41.—Older Designs of Pendulum Compressors.

The Pendulum Type.—Still another type of rotary, for the use of ammonia, is the Güttner compressor (Fig. 40). This rotary, like the Carrier centrifugal design, does not have contact between the metals of the rotor and the casing, but depends on a film of oil to maintain tightness in the clearance of about 0.0015 in. It is built to operate at 800 to 1500 r.p.m. in 1.0-ton sizes, but it does not depend on the speed for the compression as the action is positive. There is no regular suction valve although a suction check valve is used, but a plate disc is employed as a discharge valve. Tests[10] indicate that the volumetric efficiency is high, but that

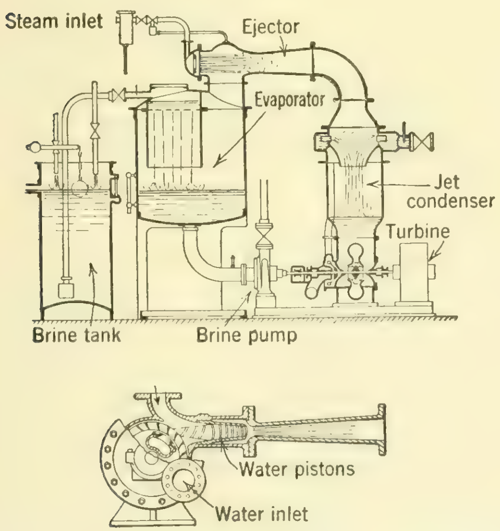


Fig. 42.—The Westinghouse-Leblanc Refrigerating Machine.

[10] H. J. Macintire, A. S. R. E. Journal, March, 1925, Plank, Krause, & Tomm, V. D. I., 1925.

the horse power per ton, as in many small machines, is also high. The older designs operating on this principle are shown in Fig. 41.

The Water Vapor Refrigerating Machine.—The water vapor refrigerating machine (the Westinghouse-Leblanc) has been proposed but has not been developed to any extent in the United States. It is a " safety " refrigerating system but it needs a high vacuum and is limited to rather high refrigerating temperatures, such as in the case of air cooling down to 50 or 40 deg. F. The high vacuums are best obtained by means of " water pistons " (Fig. 42) combined with steam ejector nozzles.

In the ejector type of water vapor refrigerating machine the underlying principle is that of a mass in motion communicating motion to another mass. The resulting velocity of the mixture will be expressed by the relation:

$$\frac{mv^2}{MV^2} = \frac{mM}{(m + M)^2}$$

If

$$\frac{M}{m} = y,$$

then

$y =$	1	2	3	4	5
$\frac{mM}{(m+M)^2} =$	0.25	0.222	0.187	0.160	0.139

where M and m are the two masses, V = the velocity of 1st mass and v = the velocity of 2nd mass. Calculation shows that the amount of ejector steam required is 3.7 lb. at 25 deg. F., 2.7 lb. at 35 deg. F. and 2.2 lb. at 45 deg. F. brine temperature per pound of " steam " boiled out of the evaporator.

THE MULTIPLE EFFECT COMPRESSOR (Dual Compression)

The multiple effect compressor of Gardiner T. Voorhees is an attempt to eliminate one compressor when two evaporating temperatures are required in a plant. For reasons of high capacity and economy as regards power input it is desirable to operate a compressor at as high a suction pressure as possible, and in the case of the two temperature plant when operated with only one compressor [11] the suction pressure carried must be that for the lower temperature room. The result of this is that the other load is carried at a condition of greater piston displacement and greater power consumed than would be required under

[11] One side of the double-acting, and one cylinder of the twin vertical single-acting compressor can be used for the low temperature suction if desired.

more advantageous operating conditions. In the Voorhees device the ordinary suction stroke is taken, drawing gas from the colder evaporating coils (the lower suction pressure) and at or near the end of the stroke either ports in the cylinder will be uncovered or certain valves be operated mechanically (Fig. 46) so as to admit gas from the other coils which are operated at the higher boiling pressure.

The action of the device for securing a multiple effect compression is shown on the indicator card in Fig. 43 as well as the action of admitting the heavier pressure gas theoretically at constant volume. However, as

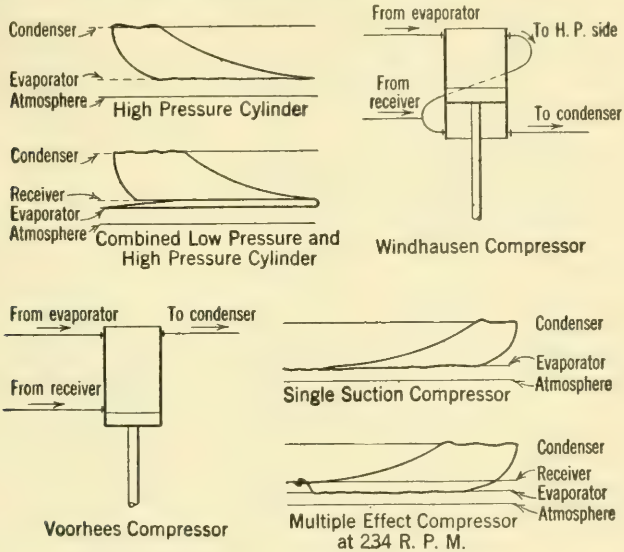


FIG. 43.—Dual (2 pressure) Compression.

the compression speed increases, this constant volume process is only approximated. The Windhausen method (Fig. 43) is not really multiple effect compression, but is more nearly stage compression. The Voorhees method is used by the Apeldoorn Company of Holland, the Seagers Company of England, and—to some slight extent—the Carbondale Machine Company and the Howe Company in the United States. As a part of the carbon dioxide compression it is of considerable value because of certain inherent failings of this refrigerant—particularly in respect to the heat of the liquid which is a large proportion of the thermal potential of the dry saturated vapor. In this case there is considerable value in permitting the refrigerant to operate under, first, a receiver pressure and, finally, the evaporating coil's pressure (Fig. 44). During the first drop of pressure, a certain amount of gas is evolved, due to the

cooling of the liquid, and this gas may be returned to the compressor at the receiver pressure instead of passing on to the evaporating coils without adequate return. For general refrigeration the multiple effect compression has not been popular because of the difficulty in adjusting the loads. In order to show the advantage to be derived *theoretically* the following problem will be solved:

Problem.—The condenser pressure is 185 lb. per square inch abs., suction pressure 30 and 45 lb. abs., cylinder capacity 10 cu. ft., saturated gas assumed at the beginning of the compression, no clearance in the cylinder and adiabatic compression. It is assumed that the compressor will operate under 30 lb. suction pressure in the first case, and with multiple effect in the second case. The problem

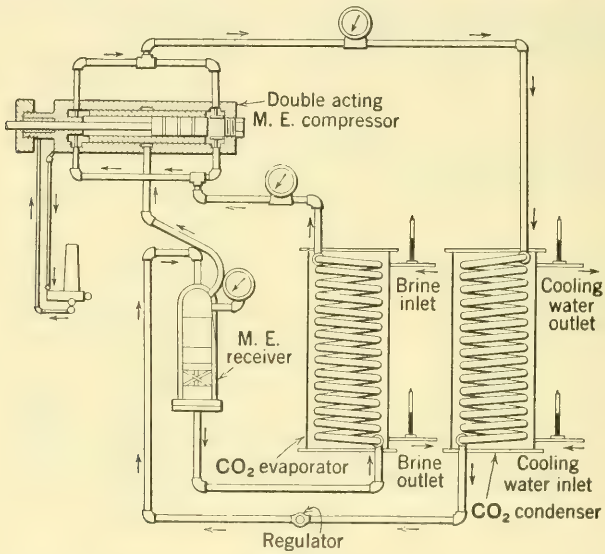


FIG. 44.—Dual Compression Applied to Carbon Dioxide Refrigeration.

is to find the percentage increase in the refrigerating capacity and the percentage increase in the power requirements for the second case. The refrigerating effect under 30 lb. suction is,

$$i''_{30\text{ lb.}} - i'_{185\text{ lb.}} = 611.4 - 145.2 = 466.2 \text{ B.t.u.}$$

$$i''_{45\text{ lb.}} - i'_{185\text{ lb.}} = 616.9 - 145.2 = 471.7 \text{ B.t.u.}$$

The weight of ammonia in 10 cu. ft. at 30 lb. abs. pressure is $\frac{10}{9.23} = 1.08$ lb.

The weight of ammonia in 10 cu. ft. at 45 lb. abs. pressure is $\frac{10}{6.3} = 1.59$ lb.

The difference, $1.59 - 1.08 = 0.51$ lb. is the amount (theoretically) of the ammonia that enters the cylinder at constant volume at the end of the suction stroke,

not making any allowance for the wire drawing which would be expected during the passage of the gas from the evaporating coils to the cylinder, and the difference in the temperatures of the two vapors. As the compressor is operated in both conditions at the same speed and the piston displacement in each case is identical, the mean effective pressure may be used to compare the power input. In the single case the refrigerating effect is the product of $1.08 \times 466.2 = 503$ B.t.u. and the mean effective pressure is 67.7 lb. per square inch. In the second case the refrigeration is $(1.08 \times 466.2) + (0.51 \times 471.7) = 743.5$ B.t.u. and the mean effective pressure is 90 lb. per square inch.

The relative advantage, then, works out as follows:

$$\text{Multiple effect compression : regular compression} = 743.7 : 503 = 1.47$$

as far as the capacity is concerned, and the power input is as $90.0 : 67.7 = 1.33$. This shows that a *gain* in the capacity of about 47 per cent is accomplished by the use of the multiple effect compression device, incidental with an increase of 33 per cent in power. This is the maximum advantage and in practice it is hardly to be expected that the full theoretical increase will be realized. If two machines had been used instead of the single two-effect compressor, one operating at 30 lb. and the other at 45 lb. suction, there would have been approximately twice the friction developed. It would seem that there are some theoretical advantages in general refrigerating practice which might disappear entirely when the conditions of operation are not the best.

In carbon dioxide compression the following problem will be of interest:

Given: The compressor is $5\frac{3}{4}$ in. by 18 in. to operate at 150 r.p.m. The piston rod is $1\frac{3}{4}$ in. and the compressor is to maintain a 25-deg. and a 40-deg. F. evaporating temperatures with 85 degrees liquefaction temperature. Taking the clearance as $4\frac{1}{2}$ per cent, the multiple effect in the cylinder as 0.625 in. diameter and the volumetric efficiency as 80 per cent, the gas entering the cylinder per minute is 59.7 cu. ft. or $\frac{59.7}{0.188}$ lb. CO_2. Taking the volume of the clearance space as 3.48 cu. ft., the temperature of the clearance gas as 142 deg. F. and the specific volume of this gas as 0.096 then the weight of the clearance gas is $\frac{3.48}{0.096} = 36$ lb. The total weight of gas becomes $36 + 317 = 353$ lb. of gas in the cylinder at the end of the low-pressure suction.

Taking the specific volume of the gas at 583 lb. and 48.5 deg. F. as 0.152 cu. ft. the volume occupied by this gas will be $353 \times 0.152 = 53.7$ cu. ft. The cylinder volume is 81.0 and therefore the amount of gas entering through the ports is 27.3 cu. ft. theoretically or $0.95 \times 27.3 = 25.9$ cu. ft. The weight of gas entering through the ports is $\frac{25.9}{0.152} = 170.5$ lb. per minute. Taking the net refrigeration at 25 deg. F. at 56.9 B.t.u. per pound the total refrigeration, at the low pressure, is

$$56.9 \times 317 = 18{,}000 \text{ B.t.u. per minute} = 90 \text{ tons refrigeration.}$$

Taking the net refrigeration at the higher pressure at 55.0 B.t.u. the total refrigerais $170.5 \times 55.0 = 9400$ B.t.u. $= 47$ tons of refrigeration.

When the multiple effect is used to cool the liquid the problem is not so straightforward. In this case it is necessary to assume an intermediate evaporating temperature, and then calculate the volume obtained which must be enough for the

cylinder dimensions. Several calculations are usually necessary before the proper pressure is obtained.

STAGE AMMONIA COMPRESSION

Stage ammonia compression is not a recent idea, but its real development did not start until about 1915. At that time the Central Cold Storage Company and the Beatrice Creamery (both of Chicago) and the Ninth Street Terminal Warehouse of Cleveland, of 250, 200 and 150 tons respectively were installed using two-stage compression for refrigerating temperatures between − 10 and − 20 deg. F. Since that time progress in stage compression has been very rapid, and its advantages have been brought out by papers presented before the technical societies by H. Sloan,[12] George Horne[13] and Thomas Shipley.[14] As a result it is conceded (1925) that stage compression is of advantage if the ratio of the compression $\left(\frac{p_2}{p_1}\right)$ is large enough, and as a rule that this condition prevails when the suction pressure is lowered to about 5 lb. gage. To the contrary the usual ice-making installation does not require stage compression, unless the condenser pressure is high.

Previous to 1915 the first attempt at stage compression was that of Norman Selfe who secured a patent in 1880. He used a form of trunk piston for the high-pressure piston; the design including a vertical single-acting cylinder where the compressed gas *was not* cooled between the stages. The St. Clair two-stage compressor was built later on in the eighties (Ice and Refrigeration, Jan., 1894) where water cooling was made use of between the stages. About 1890 Linde developed his two-stage compressor which also was water cooled, and his machine was used successfully in Europe and Australia. Then interest seemed to disappear because it was not clear that stage compression offered sufficient advantages to offset the many disadvantages due to increased fixed charges and the more complicated compressor. When there was a demand for low-temperature refrigeration the absorption machine appeared to be able to carry this kind of load with satisfaction and economy of performance, especially as steam was used universally to provide power for the refrigerating machine up to about 1915.

Occasionally the *booster compressor* was used when a relatively small part of the load required a low temperature. These booster compressors took the vapor from the low-pressure coils and raised the pressure

[12] H. Sloan, Amer. Soc. Refrigerating Eng., Dec., 1916.

[13] George Horne, Amer. Soc. Refrigerating Eng., May, 1922.

[14] Thomas Shipley, National Assoc. Practical Refrig. Eng., Nov., 1924.

to that of the other coils so that the standard compressor could handle all of the gas and discharge it all into the condenser. The booster was very convenient, and served its purpose in a satisfactory manner. However, stage ammonia compression is a problem by itself and much more complicated than in the case of the compression of air.

Stage compression for ammonia is different from that of air because

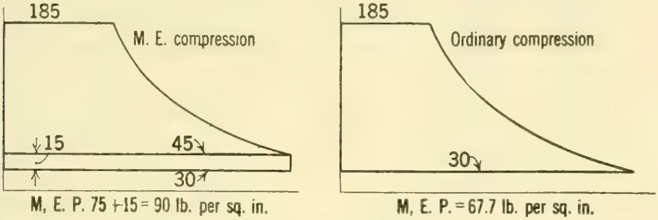


FIG. 45.—Theoretical Indicator Diagrams for Dual Compression.

ammonia in a part of the cycle is a liquid, and in addition the temperatures carried in part of the cycle are frequently 100 degrees below the temperature of the atmosphere. Also, unlike air compression, the pressure range varies very decidedly with the seasons for in the winter time the pressure ratio may be as small as 6.0 whereas in the summer this ratio of the discharge to the suction pressure may be increased to 12.0. The air compressor is likely to be operated for the same discharge pressures summer and winter.

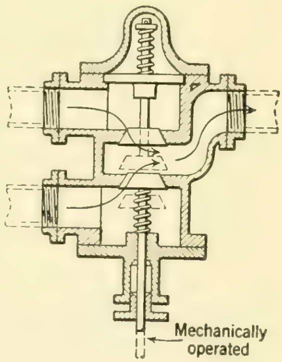


FIG. 46.—Dual Compression Mechanically Operated Valve.

In the refrigerating cycle there is an evolution of gas during the pressure drop incidental to the passage through the expansion valves. This gas is formed by the vaporization of some of the liquid ammonia and may amount to from 6 to 15 per cent of the liquid passing the valve, the amount depending on the initial and the final temperatures of the liquid during the passage of the expansion valve. In stage ammonia compression it has been the usual custom to use two *pressure-reducing* valves, and to permit the gas evolved during the first pressure drop to be separated from the liquid and to pass immediately into the suction line to the high-pressure cylinder. There are two reasons for this method; first the gas has a negligible amount of value as a refrigerating medium, and secondly, permitting it to pass into the low-pressure coils increases in proportion

the amount of gas to be handled by the low-pressure cylinder, and increases also the work performed by the compressor. The specific volume of ammonia assumes very large proportions as the pressure is reduced, and in the following problem the specific volume (the volume of one pound) is 16.66 and the per cent of the ammonia gasified during the first pressure reduction is 9.16 per cent—point *8* in the figure. If this gas is allowed to get into the low-pressure coils then the low-pressure cylinder would have to pump it up again to the intermediate pressure, the pressure of 53.7 lb. in the problem, and there would be little useful refrigeration to offset the expense of the process. The work done in the high-pressure cylinder would be the same in every case except for the superheat in the gas entering the high-pressure cylinder, which affects both the volume of the fluid to be compressed and therefore the amount of the work done in the cylinder.

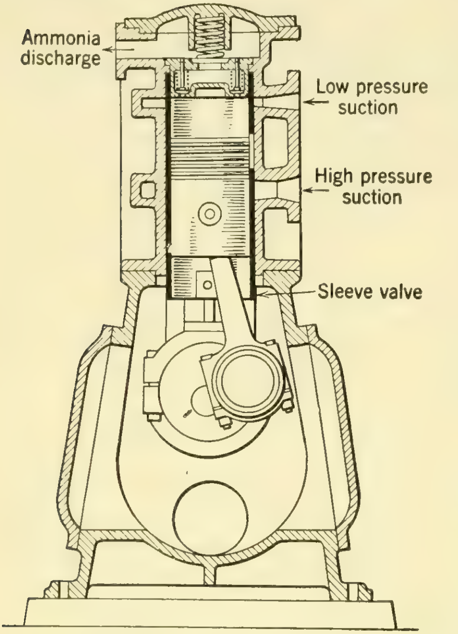


FIG. 47.—The West Sleeve Type [mechanically operated] Ammonia Compressor.

Different cycles of operation are used in stage compression depending on the method of handling the gas after it is discharged from the low-pressure cylinder. The air compressor—dealing as it does with temperatures above that of the atmosphere—attempts to cool the discharged gas from the low pressure to the initial temperature at suction. This is very easily done by the use of water-cooling coils. In ammonia compression, in the illustrative problem, the temperature of saturation at the intermediate pressure is 25 deg. F. which is considerably cooler than any water which would be available for the purpose of intermediate cooling. In fact, if cooling down to 25 degrees is required, the only way that it can be done is to make use of ammonia, but ammonia at the pressure of the low pressure, giving − 25 deg. F. boiling temperature, would not be an economical medium of operation, and one would not think for a moment of trying to cool this gas down to − 25 degrees as would be done in air compression.

It is clear therefore that cooling back to the isothermal is out of the question for ammonia compression. The very best that can be done in the intercooler is to cool down to the temperature of saturation at the intercooler pressure, and this may be done by (*a*) cooling entirely by means of liquid ammonia, by permitting the gas discharged from the low-pressure cylinder to pass into a sort of accumulator where the superheated gas and the liquid refrigerant will mix and by (*b*) cooling the superheated gas by water to (say) 70 deg. F. and by allowing this gas now superheated 45 degrees in the problem to pass into a second cooler where liquid ammonia at 25 degrees is held. The superheated gas will

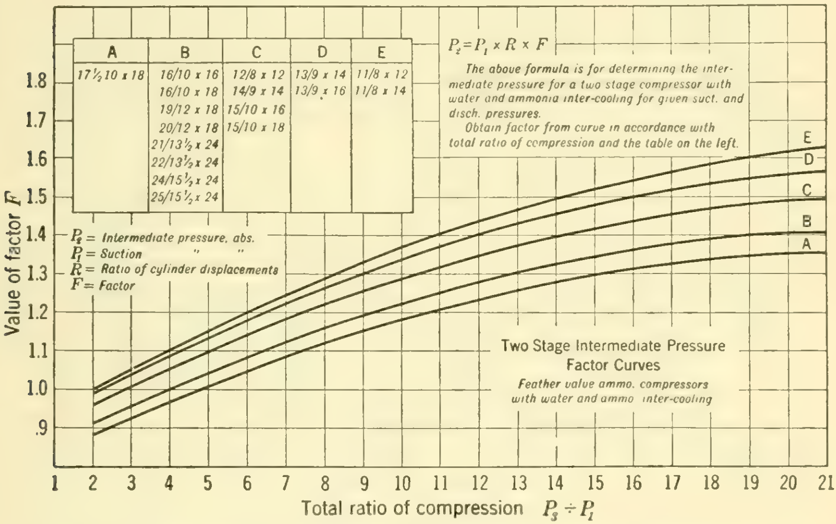


Fig. 48.—Two-stage Compression; Intermediate Pressure Determination.

be cooled, and some of the liquid will be evaporated as it absorbs heat and boils at 25 deg. F. The illustrative problem also assumes a case (*c*) where the second process (*b*) is omitted and gas superheated 45 degrees is permitted to pass on to the high-pressure cylinder.

Lately there has been a tendency on the part of a few persons to try to simplify the stage compression cycle. This has been directed against the use of the accumulator (the intercooler) and towards the use of only one expansion valve with cooling between the stages by water only. In the illustrative example the temperature after compression in the low-pressure cylinder up to 53.7 lb. is about 107 deg. F. and cooling in the aftercooler with the use of water would be from 107 to 70 degrees. This temperature of the gas entering the high-pressure cylinder would not be excessive provided the condenser pressure was not very high. In the illustrative example (Fig. 49) it will be seen

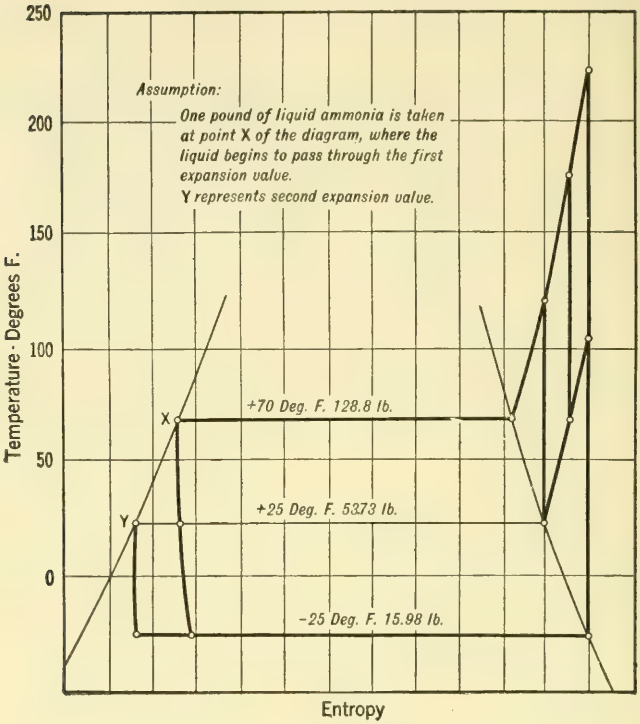


FIG. 49.—Simple and Stage Compression.

Problem	Case A Cool to + 25 Deg. F. with Liquid NH_3	Case B Cool to 70 Deg. F. with Water and to + 25 Deg. F. with NH_3	Case C Cool to 70 Deg. F. with Water	Case D Simple Compression	Case E No Liquid Intercooling Gas Cooled to 70 Deg. F. with Water
Weight of liquid evaporated in cooling l.p. disch. gas, lb.	0.0725	0.04275			
Weight liquid evaporated in first exp. valve, lb.	0.0916	0.0916	0.0916	0.178	0.0916
Weight of liquid passing to second expansion, lb.	0.8359	0.8657	0.9084		
Weight ammonia as liquid available for refrigeration, lb.	0.7588	0.7860	0.822	0.822	0.822
Refrigerating effect, B.t.u.	445.50	461.0	484.2	482.7	482.7
Work done—l.p.cyl. ft., lb.	44,450	46,020	48,300	....	53,150
Work done—h.p. cyl. ft.-lb.	39,800	39,800	44,450	....	44,450
Total work of compression, ft.-lb.	84,250	85,820	92,750	101,400	97,600
Probable vol. eff. h.p. cyl.	0.9233	0.9233	0.9165	0.8110	0.9165
Probable eff. l.p. cyl.	0.9008	0.9008	0.9008	....	0.9008
Total work—allowing for vol. eff. ft.-lb.	92,450	94,250	102,100	125,000	107,500
Coefficient of performance	3.75	3.81	3.690	3.00	3.49
Hp. per ton of refrigeration	1.257	1.238	1.277	1.568	1.352

Comparative Efficiencies of Simple and Compound Compression.

that this case—Case E—is not so good as far as the horse power per ton of refrigeration is concerned. Referring to Cases A, B and C it will be noticed that there is slight difference in this respect, but that in Case B the lowest horse power per ton of refrigeration is found as developed from the theoretical calculation. Finally there are given in the table values calculated for simple compression, Case D. All the calculations are based on the premise of having one pound of liquid at 70 deg. F. at the expansion valve.

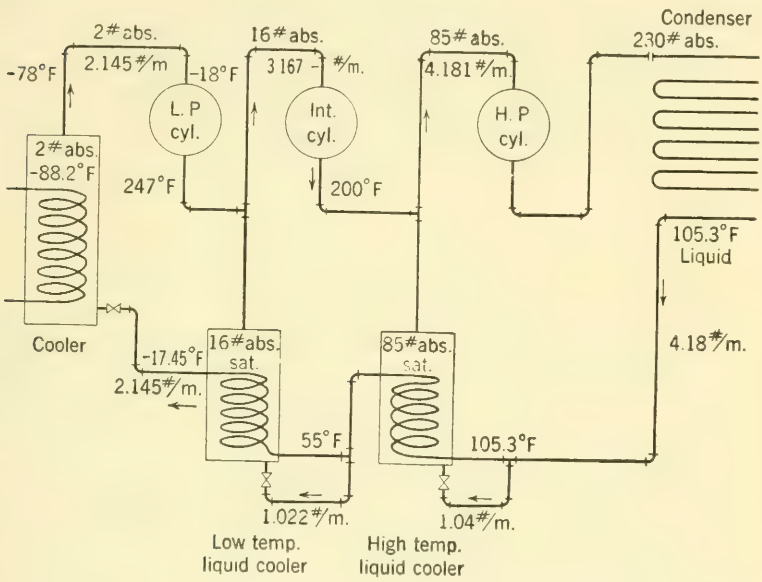


FIG. 50.—Ammonia Compression in Three Stages.

In general it can be said that the best arrangement is usually the simplest, but this is not the case when dealing with low-temperature refrigeration. Ordinarily there is no advantage in cooling the refrigerant by means of itself, although that is what is done in the case when using the accumulator in the flooded system. Referring to the table it will be seen that the simplest cycle for stage compression, Case D, is not the best one, as far as the power requirements are concerned, by any means, although it has one less piece of equipment, the accumulator. With the accumulator and cooling only to 70 deg. F. with water the horse power per ton of refrigeration is 1.277 as compared with 1.352 in the

cycle in Case E. The best results are given in Case B which has the accumulator, two expansion valves and has cooling with water to 70 degrees and with ammonia to 25 degrees.

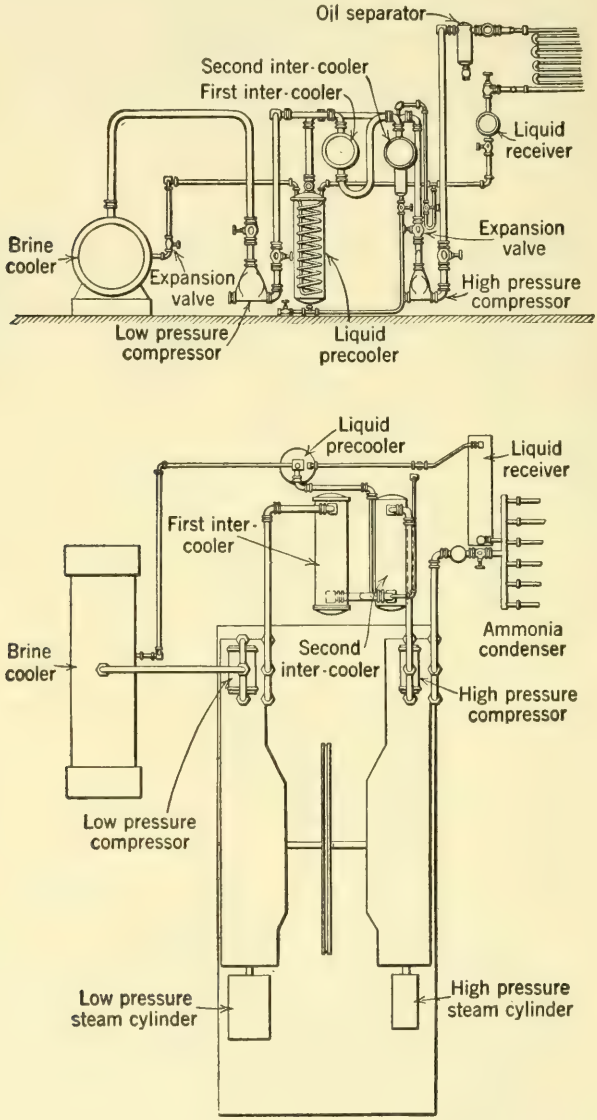


Fig. 51.—Two-stage Ammonia Compressor.

Ammonia compression in three stages is rare at the present time, as only extremely severe operating conditions will warrant such a design.

A Carbondale three-stage compressor (Fig. 50) installed in one of the rubber manufacturing companies had a 14 and 8 and 4 by 12 inch stroke,

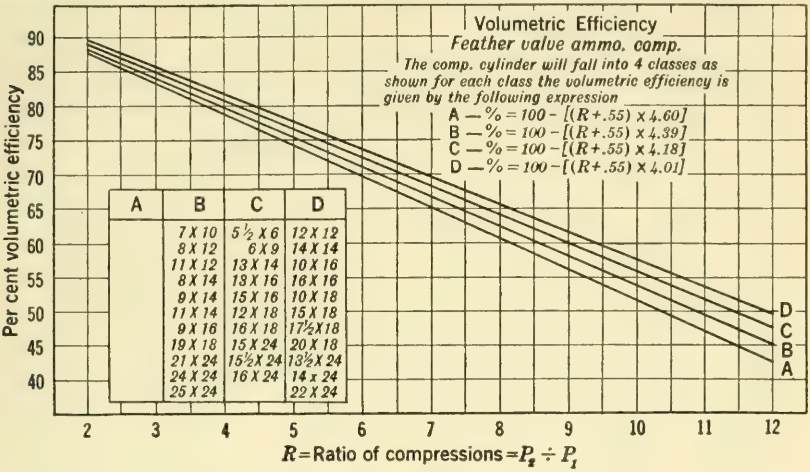


FIG. 52.—Volumetric Efficiency.

single-acting except for the low pressure (14 by 12) cylinder which was double-acting. The total piston displacement was 570 cu. ft. per minute at 225 r.p.m. and the operating conditions were for 2.0 lb. suction and

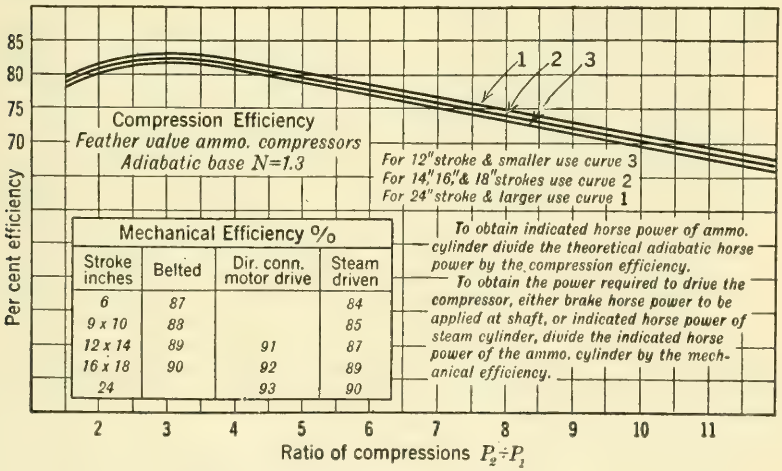


FIG. 53.—Compression Efficiency.

230 lb. condenser pressure. This machine was designed for 6.0 tons under good conditions of operation and would require about 30 brake horse power.

Compressor Details.—Tables 5 to 13 give approximate sizes of standard ammonia and carbonic compressors built by various manufacturers of refrigerating machines. In addition Figs. 52, 53 and 54 give the volumetric efficiency, cubic feet per ton and horse power per ton used by a well-known manufacturer. Every manufacturer has curves similar to these last, and the only variations are in the volumetric efficiency and the allowance for the leakage through the valves and piston rings and in the imperfections of operation. In Figs. 29 and 30, for example, no allowance was made for mechanical defects nor for the personal equation of the operator so that where the values given in Figs. 29 and 30 are used it is necessary to make some allowance as a factor of safety. Likewise the horse power per ton of refrigeration as given in Figs. 31 and 32 are smaller than would be specified in practice. In starting up, especially in automatic installations, the load is considerably greater than under usual running conditions because of the high suction pressures, and so the result is that a larger motor is installed than would be indicated from the consideration of Figs. 31 and 32. Tables 3 to 14 are taken from manufacturer's data without editing.

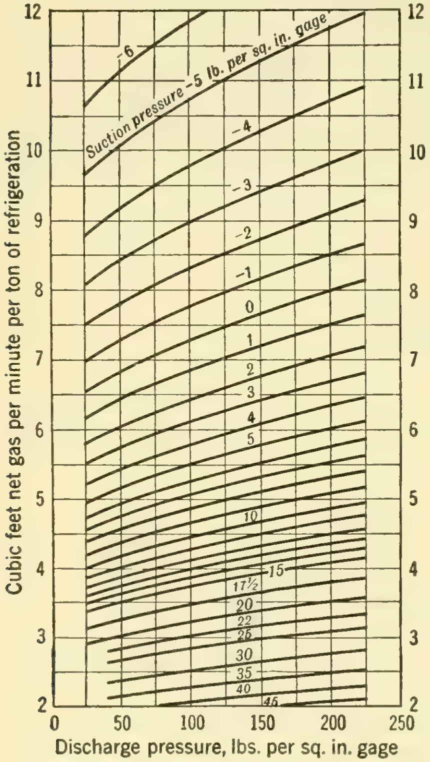
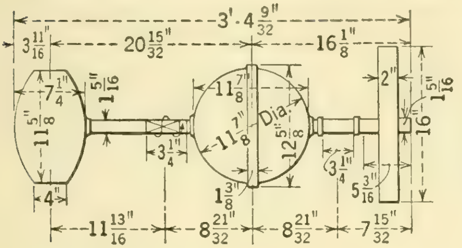


FIG. 54.—Piston Displacement per Ton of Refrigeration per Minute.

In the tables the floor space and in a number of cases the overall height are given. Piping connections are given in Chapter IV, as are also the details on suction traps and oil separators. Some electric motor characteristics are shown in Chapter XXII, and some steam engine details are collected in Chapter XXIII.

TABLE 2

Characteristics of No. 2 Audiffren-Singrun Refrigerating Machine



Direction of rotation is clockwise when facing machine at pulley end

DUMBBELL AND PULLEY

Dumbbell Type 2E

Keyway in shaft.... $\frac{1}{4}$ in. by $\frac{1}{8}$ in. by $4\frac{3}{4}$ in.

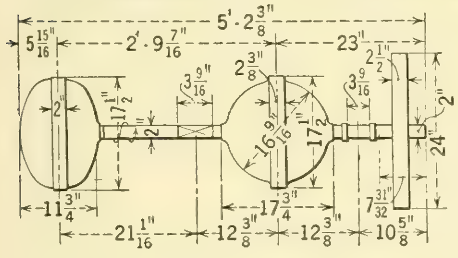
Weight.......................... 140 lb.

Weight crated.................... 200 lb.

R.p.m.............................. 380

Condensing Water Temperature at Overflow		50°	59°	68°	77°	86°	95°
Capacity in B.t.u. absorbed per hour at brine temperature of..............	50°	4200	4000	3800	3640	3450	3250
	41°	3560	3400	3250	3090	2930	2730
	37°	3290	3170	3010	2850	2690	2530
	32°	3050	2890	2770	2610	2490	2340
	27°	2610	2500	2380	2260	2140	2020
	23°	2410	2340	2220	2100	1980	1860
	14°	1940	1820	1740	1660	1580	1460
Capacity in pounds of ice per hour with water for cans at 59 degrees and brine at 27 degrees......................	....	13	12	11	$10\frac{1}{2}$	10	9
Condensing water required per hour in gallons with temperature rise of....	9°	57	53	49	45	41	38
	18°	28.5	26.5	24.5	22.5	20.5	19
	27°	....	18	16.5	15	13.5	13
	36°	....	....	12.5	11.5	10.5	9.5
With brine at 27 deg. F................							
Horse power required brine at $27\frac{1}{2}$ deg. F	....	.48	.50	.53	.55	.58	.60

Characteristics of No. 3 Audiffren-Singrun Refrigerating Machine



Direction of rotation is clockwise when facing machine at pulley end

DUMBBELL AND PULLEY

Dumbbell Type 3E

Keyway in shaft.... $\frac{3}{8}$ in. by $\frac{3}{16}$ in. by $7\frac{5}{16}$ in.

Weight.......................... 390 lb.

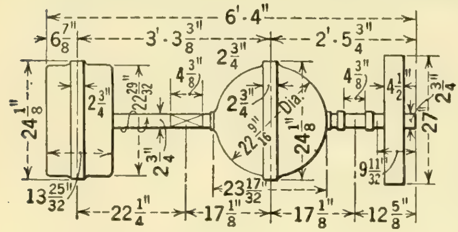
Weight crated.................... 490 lb.

R.p.m.............................. 280

Condensing Water Temperature at Overflow		50°	59°	68°	77°	86°	95°
Capacity in B.t.u. absorbed per hour at brine temperature of..............	50°	10,500	10,000	9500	9100	8610	8120
	41°	8,910	8,510	8120	7720	7320	6830
	37°	8,220	7,920	7520	7120	6730	6340
	32°	7,620	7,230	6930	6530	6240	5840
	27°	6,530	6,240	5940	5640	5340	5040
	23°	6,040	5,840	5540	5250	4950	4650
	14°	4,850	4,550	4350	4160	3960	3860
Capacity in pounds of ice per hour with water for cans at 59 degrees and brine at 27 degrees......................	....	33	30	27	26	24	23
Condensing water required per hour in gallons with temperature rise of....	9°	120	114	107	100	93	85
	18°	60	57	54	50	47	43
	27°	....	38	36	33	31	28
	36°	....	....	27	25	24	22
With brine at 27 deg. F................							
Horse power required brine at $27\frac{1}{2}$ deg. F	.	.90	1.02	1.14	1.26	1.38	1.50

TABLE 2—*Continued*

Characteristics of No. 4 Audiffren-Singrun Refrigerating Machine



Direction of rotation is clockwise when facing machine at pulley end

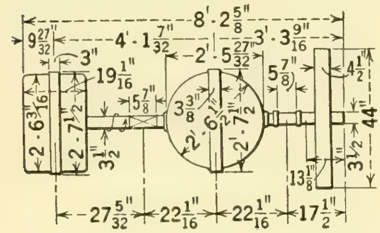
DUMBBELL AND PULLEY

Dumbbell Type 4H

Keyway in shaft.....½ in. by ¼ in. by 8¾ in.
Weight..........................850 lb.
Weight crated...................1085 lb.
R.p.m...........................190

Condensing Water Temperature at Overflow		50°	59°	68°	77°	86°	95°
Capacity in B.t.u. absorbed per hour at brine temperature of..............	50°	21,000	20,000	19,000	18,200	17,220	16,240
	41°	17,820	17,020	16,240	15,440	14,640	13,660
	37°	16,440	15,840	15,040	14,240	13,460	12,680
	32°	15,240	14,460	13,860	13,060	12,480	11,680
	27°	13,060	12,480	11,880	11,280	10,680	10,100
	23°	12,080	11,680	11,080	10,500	9,900	9,300
	14°	9,700	9,100	8,700	8,320	7,920	7,720
Capacity in pounds of ice per hour with water for cans at 59 degrees and brine at 27 degrees......................	....	66	60	55	52	49	45
Condensing water required per hour in gallons with temperature rise of....	9°	230	220	210	195	185	175
	18°	115	110	105	97	93	88
	27°	....	74	70	65	62	58
	36°	....	....	53	49	47	44
With brine at 27 deg. F.................							
Horse power required brine at 27½ deg. F.	....	1.6	1.8	2.05	2.15	2.3	2.5

Characteristics of No. 6 Audiffren-Singrun Refrigerating Machine



Direction of rotation is clockwise when facing machine at pulley end

DUMBBELL AND PULLEY

Dumbbell Type 6H

Keyway in shaft......½ in. by ¼ in. by 12 27/32 in.
Weight..........................1700 lb.
Weight crated...................2100 lb.
R.p.m...........................140

Condensing Water Temperature at Overflow		50°	59°	68°	77°	86°	95°
Capacity in B.t.u. absorbed per hour at brine temperature of..............	50°	42,000	40,000	38,000	36,400	34,490	32,480
	41°	35,640	34,040	32,480	30,880	29,280	27,320
	37°	32,880	31,680	30,080	28,480	26,820	25,360
	32°	30,480	28,920	27,720	26,120	24,960	23,360
	27°	26,120	24,960	23,760	22,560	21,360	20,200
	23°	24,160	23,360	22,160	21,000	19,800	18,600
	14°	19,400	18,200	17,400	16,640	15,840	15,440
Capacity in pounds of ice per hour with water for cans at 59 degrees and brine at 27 degrees......................		132	120	110	105	99	92
Condensing water required per hour in gallons with temperature rise of....	9°	420	410	400	390	380	370
	18°	210	205	200	195	190	185
	27°	....	137	134	130	127	124
	36°	....	....	100	95	93	92
With brine at 27 deg. F.................							
Horse power required brine at 27½ deg. F.	....	3.6	3.9	4.1	4.3	4.6	4.8

TABLE 3

TRANSMISSION DATA FOR BELT-DRIVEN V. S. A. ENCLOSED TYPE COMPRESSORS

Size of Machine, Inches	Standard Speed, R.p.m.	Diameter of Belt Wheel, Inches	Face of Belt Wheel (Not Including Pockets), Inches	Weight of Belt Wheel, Pounds	Motor, Full Load, R.p.m., 60 Cycles	Diameter of Motor Pulley, Inches	Face of Motor Pulley, Inches	Width of Double Load Belt, Inches	Width of Single Load Belt, Inches	Distance, Center to Center, of Belt Wheel and Motor Pulley, Feet	Approximate Length of Belt, Feet
3× 3 (Single)	300	26	3½	180	1720	4½	3½	......	3	6	17
3× 3 (Double)	300	26	3½	180	1720	4½	3½	......	3	6	17
4× 4 (Single)	275	30	4½	300	1720	5	4½	......	4	6	18
4× 4 (Double)	275	30	4½	300	1720	5	4½	......	4	6	18
5× 5	240	36	6½	600	1150	7½	6½	......	6	8	22
6× 6	220	42	7½	600	1150	8	7½	......	7	8	24
7× 7	210	48	8½	1000	1150	9	8½	6	8	10	29
8× 8	200	50	8½	1200	870	11½	8½	8	......	10	30
9× 9	190	60	10½	1500	870	13	10½	10	......	12	34
10×10	180	72	12½	1800	870	15	12½	12	......	14	40
12×12	170	86	14½	3250	870	17	14½	14	......	16	47

Weight of belt wheels based on 225-lb. H. P., and 25-lb. B. P. Standard Speed.

Compressors requiring 20 per cent or more speed reduction will require special belt wheels of greater weight.

TABLE 4

Motor Engine Horse Power for V. S. A. Enclosed Compressors

185 Lb. Condenser Pressure

Cylinder Number	Cylinder Bore	Cylinder Stroke	Revolutions per minute	6 Lb. —15 deg. F. E.H.P.	6 Lb. —15 deg. F. M.H.P.	9 Lb. —10 deg. F. E.H.P.	9 Lb. —10 deg. F. M.H.P.	12 Lb. —5 deg. F. E.H.P.	12 Lb. —5 deg. F. M.H.P.	16 Lb. —0 deg. F. E.H.P.	16 Lb. —0 deg. F. M.H.P.
1	3	3	330	1.35	1.22	1.43	1.29	1.51	1.36	1.58	1.43
1	4	4	275	2.85	2.60	3.01	2.8	3.18	2.9	3.32	3.10
2	4	4	275	5.46	5.10	5.80	5.40	6.10	5.70	6.38	6.00
2	5	5	240	9.10	8.60	9.60	9.20	10.1	9.7	10.6	10.1
2	6	6	220	14.2	13.7	15.1	14.4	15.9	15.2	16.6	15.9
2	7	7	210	21.2	20.5	22.5	21.6	23.7	22.8	24.8	23.9
2	8	8	200	29.9	28.9	31.6	30.6	33.3	32.2	34.8	33.8
2	9	9	190	40.5	38.8	42.8	41.0	45.2	43.4	47.3	45.3
2	10	10	180	52.5	49.9	55.5	52.8	58.5	55.8	61.3	58.3
2	12	12	170	86.0	80.7	90.8	85.3	96.0	90.3	100.2	94.2
				19 Lb. 5 deg. F. E.H.P.	**19 Lb. 5 deg. F. M.H.P.**	**24 Lb. 10 deg. F. E.H.P.**	**24 Lb. 10 deg. F. M.H.P.**	**28 Lb. 15 deg. F. E.H.P.**	**28 Lb. 15 deg. F. M.H.P.**	**33 Lb. 20 deg. F. E.H.P.**	**33 Lb. 20 deg. F. M.H.P.**
1	3	3	330	1.65	1.50	1.70	1.54	1.76	1.59	1.80	1.63
1	4	4	275	3.47	3.20	3.60	3.30	3.71	3.40	3.80	3.50
2	4	4	275	6.65	6.30	6.90	6.50	7.10	6.70	7.30	6.80
2	5	5	240	11.1	10.7	11.5	11.0	11.9	11.3	12.1	11.6
2	6	6	220	17.4	16.6	18.0	17.2	18.6	17.8	19.0	18.2
2	7	7	210	25.9	25.0	26.8	25.8	27.7	26.8	28.3	27.2
2	8	8	200	36.4	35.3	37.7	36.3	38.9	37.7	39.8	38.4
2	9	9	190	48.4	47.4	51.1	48.9	52.8	50.5	54.0	51.7
2	10	10	180	64.0	61.0	66.2	63.0	68.5	65.3	70.0	66.4
2	12	12	170	105.0	98.4	108.4	102.0	112.0	105.4	114.5	107.6

Ammonia Suction Pressures and Corresponding Ammonia Temperatures, Pounds per Square Inch, Gage and Degrees Fahrenheit

E.H.P. = Indicated horse power of direct compressor steam engine required.
M.H.P. = Motor horse power required.

Multiply above Horse Power by Constants Given below to Obtain Horse Power at Different Condenser Pressures Given

Condensed Pressures, Pounds, Gage	Ammonia Suction Pressures 6 Lb.	9 Lb.	12 Lb.	16 Lb.	19 Lb.	24 Lb.	28 Lb.	33 Lb.
200	1.042	1.048	1.05	1.052	1.057	1.06	1.062	1.07
168	.85	.948	.945	.943	.94	.938	.93	.928
153	.8895	.8893	.8885	.888	.8872	.8868	.886	.8847

TABLE 4—*Continued*

Vertical Single-acting Machines

Volumetric Efficiencies of Compressors—Efficiency. Actual Cubic Feet Piston Displacement per Ton per Minute—Cubic Feet. Indicated Horse Power of Compressors per Ton of Refrigeration—Horse Power.

Compressor Pressure (Gage)	Corresponding Temperature, Degrees	Back Pressure (Gage) and Corresponding Temperature											
		7.94 Lb. —12 Degrees			10.31 Lb. —80 Degrees			12.89 Lb. —40 Degrees			15.67 Lb. —0 Degrees		
		EFF.	C.F.	H.P.	EFF.	C.F.	H.P.	EFF.	C.F.	H.P.	EFF.	C.F.	H.P.
126.20	75	79.0	6.32	1.38	80.2	5.83	1.30	81.4	5.13	1.16	82.6	4.47	1.05
139.40	80	78.5	6.50	1.48	79.7	5.89	1.39	80.8	5.21	1.25	81.9	4.55	1.13
153.18	85	78.0	6.60	1.58	79.1	5.95	1.48	80.2	5.30	1.34	81.3	4.64	1.24
167.92	90	77.5	6.70	1.69	78.6	6.05	1.59	79.6	5.40	1.46	80.6	4.72	1.34
183.65	95	77.0	6.79	1.83	78.0	6.12	1.68	79.0	5.50	1.56	80.0	4.81	1.45
200.42	100	76.5	6.93	1.96	77.5	6.20	1.82	78.4	5.60	1.67	79.4	4.90	1.56
218.28	105	76.0	7.05	2.10	76.9	6.30	1.93	77.8	5.71	1.79	78.7	4.99	1.64
237.27	110	75.5	7.18	2.22	76.4	6.40	2.04	77.2	5.80	1.90	78.1	5.10	1.74
258.70	115	75.0	7.35	2.37	75.8	6.50	2.16	76.6	5.90	2.01	77.5	5.20	1.84
275.90	120	74.5	7.50	2.52	75.3	6.63	2.31	76.0	6.02	2.13	76.8	5.30	1.95

Compressor Pressure (Gage)	Corresponding Temperature, Degrees	19.46 Lb. +5 Degrees			23.64 Lb. +10 Degrees			28.24 Lb. +15 Degrees			33.25 Lb. +20 Degrees		
		EFF.	C.F.	H.P.	EFF.	C.F.	H.P.	EFF.	C.F.	H.P.	EFF.	C.F.	H.P.
126.20	75	83.7	3.99	.96	84.8	3.49	.86	85.9	3.07	.77	86.9	2.70	.60
139.40	80	83.0	4.05	1.00	84.1	3.56	.95	85.2	3.13	.85	86.2	2.75	.76
153.18	85	82.3	4.14	1.14	83.4	3.62	1.03	84.4	3.20	.95	85.5	2.81	.85
167.92	90	81.7	4.20	1.23	82.7	3.69	1.11	83.7	3.25	1.02	84.7	2.87	.93
183.65	95	81.0	4.28	1.33	82.0	3.77	1.22	83.0	3.32	1.12	84.0	2.91	1.02
200.42	100	80.4	4.36	1.43	81.3	3.84	1.31	82.3	3.40	1.21	83.3	2.98	1.09
218.28	105	79.7	4.45	1.54	80.6	3.91	1.40	81.8	3.52	1.41	81.8	3.10	1.27
237.27	110	79.1	4.54	1.64	79.9	3.98	1.56	80.1	3.60	1.52	81.1	3.14	1.36
258.70	115	78.4	4.63	1.74	7.92	4.06	.68	79.4	3.66	1.63	80.4	3.21	1.46
275.90	120	77.8	4.70	1.84	78.5	4.15	1.80						

$$\text{Actual cubic feet per ton per minute} = \frac{\text{Theoretical cubic feet gas to produce one ton of refg.}}{\text{Volumetric efficiency}}$$

$$\text{I.hp. per ton} = \frac{\text{Plan}}{33{,}000 \times \text{Eff.}} \div \frac{\text{Plan}}{144 \times \text{Eff.} \times \text{Actual cubic feet per ton per minute}}$$

$$= \frac{P \times \text{Actual cubic feet per ton per minute}}{229}$$

TABLE 5

Size of Compressor	Full Speed, R.p.m.	Cubic Feet Piston Displacement per Minute (Full Speed)	Cubic Feet Piston Displacement per Rev. C.F.R.	Per Cent Friction	
				Steam drive	Belt drive
1–$6\frac{1}{2}\times 6$	130	15.0	.116	25	23
2–$6\frac{1}{2}\times 6$	130	30.0	.231	24	22
$7\frac{1}{2}\times 8$	120	49.1	.409	23	21
$8\frac{1}{2}\times 10$	110	72.3	.657	22	20
$9\frac{1}{2}\times 12$	100	38.5	.985	21	19
$10\frac{1}{2}\times 15$	95	142.8	1.50	20	18
$12\frac{1}{2}\times 18$	90	230.0	2.555	19	17
$13\frac{1}{2}\times 20$	85	281.5	3.312	18	16
15×24	80	392.7	4.91	17	16
$16\frac{1}{2}\times 28$	75	519.7	6.93	16	15
18×32	65	612.6	9.424	15	15
$20\frac{1}{2}\times 36$	60	825.2	13.75	15	15
$22\frac{1}{2}\times 36$	60	976.6	16.27	15	15
24×36	60	1131.0	18.85	15	15

FORMULAE

C.F. = Cu. ft. pist. displ. per ton per min. (From Table)
C.F.R. = Cu. ft. pist. displ. per rev. (From Table)
R.P.M. = Rev. per min. of compressor.
H.P. = Indicated amm. H.P. per ton of refrigeration. (From Table)
A.H.P. = Indicated ammonia H.P.
S.H.P. = Indicated Steam H.P.
B.H.P. = Belt H.P.
M.H.P. = H.P. of belted motor or eng.

1. $\text{Tons Ref.} = \dfrac{\text{C.F.R.} \times \text{R.P.M.}}{\text{C.F.}}$
2. A.H.P. = Tons ref. $\times$ H.P.
3. S.H.P. = A.H.P. $\times$ (1 + per cent "steam" friction).
4. B.H.P. = A.H.P. $\times$ (1 + per cent "belt" friction).
5. M.H.P. = B.H.P. $\times$ 110 per cent.

TABLE 6

Capacities of V. S. A. Enclosed Type Machines

Cylinder			Piston Speed per Minute		Displacement per Minute		Capacity in Tons per 24 Hrs., 185-Lb. Condensing								
Number	Bore	Stroke	R p m	Feet	Cubic inches	Cubic feet	6 lb. −15° F	9 lb. −10° F	12 lb. −5° F	16 lb. 0° F		19 lb. 5° F	24 lb. 10° F	28 lb. 15° F.	33 lb. 20° F.
							Refg.	Refg.	Refg.	Ice	Refg.	Refg.	Refg.	Refg.	Refg.
1	3	3	300	150	6,362	3.68	.47	.51	.65	.47	.75	.85	0.96	1.07	1.24
1	4	4	275	183	13,828	7.99	.98	1.12	1.35	1.0	1.6	1.78	2.0	2.2	2.6
2	4	4	275	183	27,646	15.98	1.96	2.24	2.7	2.0	3.2	3.5	4.0	4.4	5.2
2	5	5	240	200	47,124	27.3	3.34	3.84	4.6	3.3	5.3	5.97	6.8	7.7	8.85
2	6	6	220	220	74,644	43.2	5.5	6.6	7.6	5.5	8.8	10.0	11.4	12.6	14.5
2	7	7	210	245	113,146	65.6	8.5	10.2	11.9	8.5	13.6	15.5	17.4	19.6	22.4
2	8	8	200	266	160,848	93.08	12.2	14.6	16.9	12.1	19.5	22.1	24.9	28.0	32.1
2	9	9	190	285	217,570	125.8	16.6	19.2	22.9	16.6	26.4	29.8	33.8	37.9	43.5
2	10	10	180	300	282,744	163.62	22.1	26.1	30.4	21.9	35.0	38.8	44.9	50.4	57.8
2	12	12	170	340	461,448	267.04	35.1	41.9	48.6	35.0	56.0	63.4	71.6	80.3	92.4

Multiply above capacities by the constants given below to obtain capacities at different condenser pressures given.

Ammonia Suction Pressure

Condenser Pressure Pounds, Gage	6 Lb.	9 Lb.	12 Lb.	16 Lb.	19 Lb.	24 Lb.	28 Lb.	33 Lb.
200	.975	.969	.975	.977	.978	.977	.976	.977
168	1.029	1.028	1.028	1.028	1.02	1.027	1.026	1.022
153	1.046	1.049	1.05	1.05	1.043	1.05	1.042	1.042

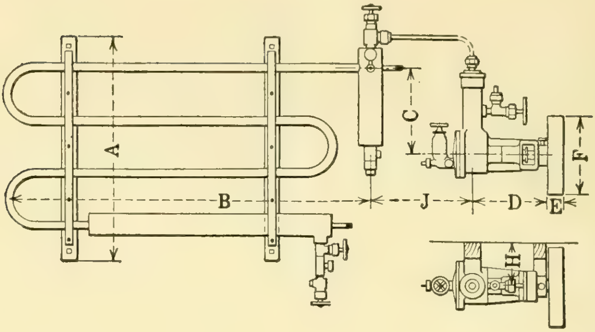


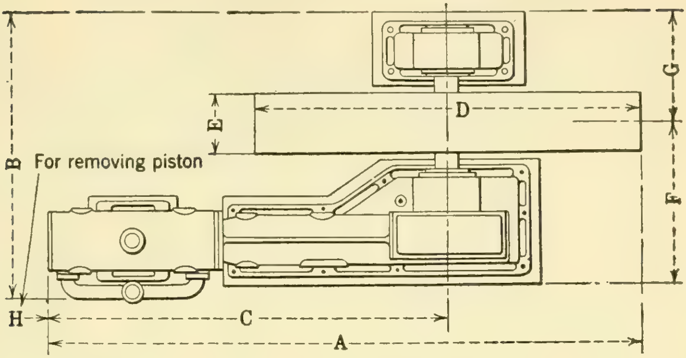
TABLE 7

Enclosed Machine, with Oil Separator, Ammonia Condenser, Receiver and Connections	Number of Machine		
	00	0	1
Heat units eliminated hourly, cooling 60 to 40 deg. F.	4000	8000	16,000
Ice-making capacity per 24 hrs. . . . tons	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{2}$
Refrigerating capacity per 24 hrs. . . . tons	$\frac{1}{4}$	$\frac{1}{2}$	1
Actual horse power required	$\frac{3}{4}$	$1\frac{1}{2}$	$2\frac{3}{4}$
Revolutions per minute	250	250	200
Shipping weight . . . . cwts.	4	6	11
Shipping measurement . . . . cu. ft.	9	16	35

Number	A	B	C	D	E	F	H	J
	Inches	Inches	Inches	Inches	Inches	Inches	Inches	Inches
00	34	53	13	$11\frac{1}{8}$	$2\frac{1}{2}$	12	$7\frac{1}{4}$	15
0	34	89	16	$14\frac{3}{8}$	3	18	10	18
1	54	114	18	15	4	21	12	22

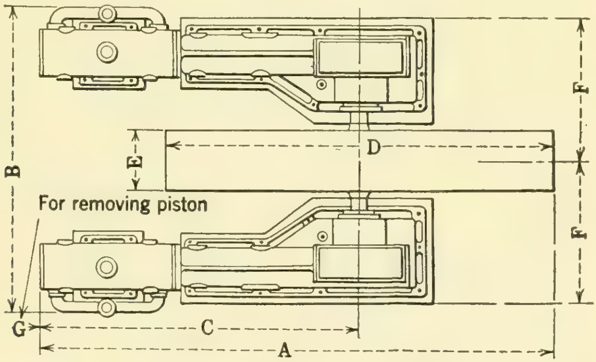
TABLE 8

Arctic Ice Machine Co. Ammonia Compressors



Cylinder Size	A		B		C		D		E	F		G		H	
Inches	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Inches	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.
9 ×13½	13	8	5	11½	9	2	9	0	12	3	5¼	2	1½	3	1
10½×15¾	15	11½	7	2¼	10	11½	10	0	14	4	0⅝	2	7⅛	3	10
12 ×18	17	3⅞	7	10 7/16	11	3⅞	12	0	15	4	4⅜	2	9½	4	6
13½×20¼	19	11 9/16	8	11 7/16	12	5 9/16	12	0	22	4	11	3	1	4	9
15 ×22½	22	2	9	2	14	8	15	0	24	5	5	3	7½	5	0

Floor space required for single-cylinder, belted compressors

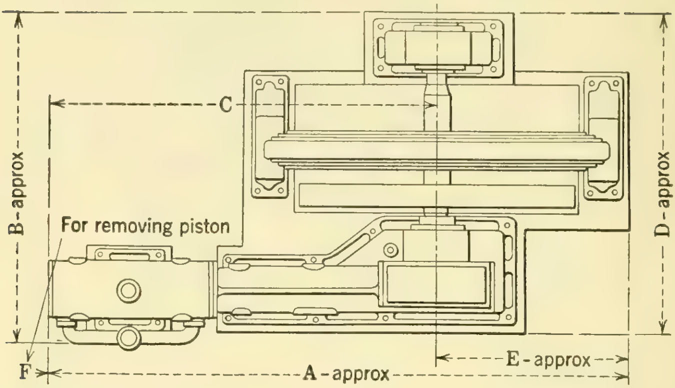


Cylinder Size	A		B		C		D		E	F		G	
Inches	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Inches	Ft.	Ins.	Ft.	Ins.
9 ×13½	13	8	8	8	9	2	9	0	22	3	11¼	3	1
10½×15¾	15	11½	9	2¼	10	11½	10	0	26	4	6⅝	3	10
12 ×18	17	3⅞	10	9⅞	11	3⅞	12	0	28	4	10⅞	4	6
13½×20¼	19	11 9/16	11	3⅞	12	5 9/16	15	0	28	5	2	4	9
15 ×22½	22	2	13	2	14	8	15	0	40	6	1	5	0

Floor space required for twin-cylinder, belted compressors

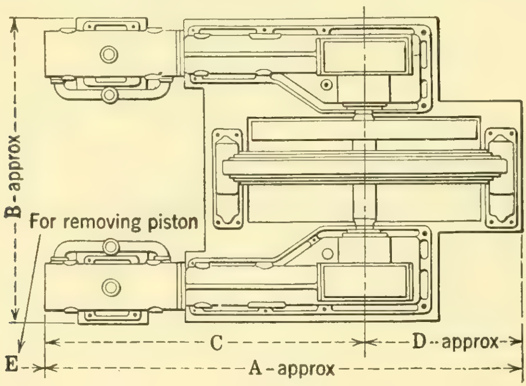
TABLE 9

ARCTIC ICE MACHINE AMMONIA COMPRESSORS



Cylinder Size	Speed	A		B		C		D		E		F	
Inches		Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.
9 × 13½	100–140	14	8	8	2	9	2	7	10	5	6	3	1
10½ × 15¾	100–140	16	7½	9	0	10	11½	8	6	5	8	3	10
12 × 18	90–130	18	0	11	0	11	3⅞	10	10	6	8⅛	4	6
13½ × 20¼	80–125	19	3	11	3	12	5 7/16	11	0	6	9 9/16	4	9
15 × 22½	75–120	22	5	13	0	14	8	12	4	7	9	5	0

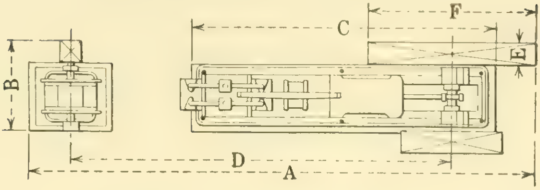
Floor space required for Single-cylinder Compressors direct-connected to synchronous motors



Cylinder Size	A		B		C		D		E	
Inches	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.
9 × 13½	14	8	9	1½	9	2	5	6	3	1
10½ × 15¾	16	7½	9	11¼	10	11½	5	8	3	10
12 × 18	18	0	13	2	11	3⅞	6	8⅛	4	6
13½ × 20¼	19	3	13	6	12	5 7/16	6	9 9/16	4	9
15 × 22½	22	5	14	4	14	8	7	9	5	0

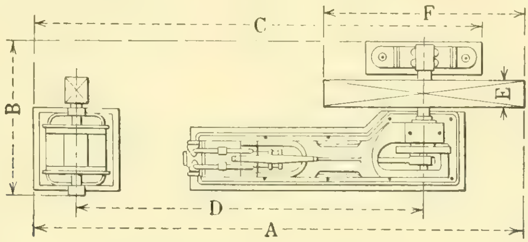
Floor space required for Twin-cylinder Compressors direct-connected to synchronous motors

TABLE 10
American Carbonic CO_2 Compressors



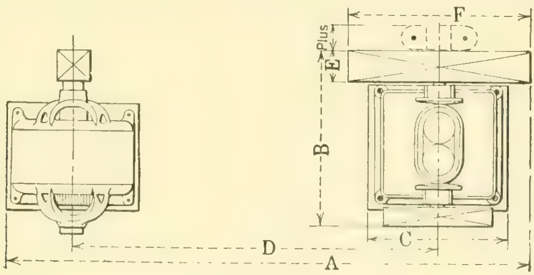
Horizontal Center Crank Type
Refrigerating Capacity, 3 to 10 Tons

Size in Tons, Refrigerating Capacity	A		B		C		D		E	F		Overall Height	
	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Inches	Ft.	Ins.	Ft.	Ins.
3 to 5	11	0	2	6	4	10½	8	0	6½	3	0	4	4
6 to 8	14	3	2	11	6	9¾	10	0	7½	5	0	5	6
8 to 10	14	6	2	11	6	10½	10	0	8½	5	0	5	6



Horizontal Side Crank Type
Refrigerating Capacity, 10 to 150 Tons

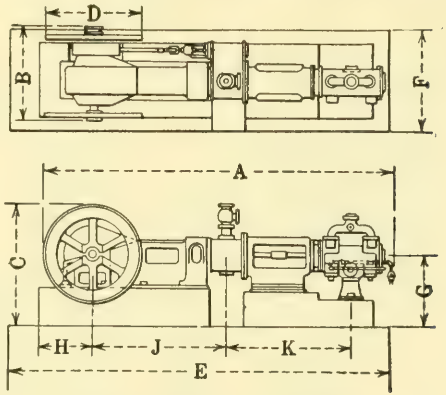
Size in Tons, Refrigerating Capacity	A		B		C		D		E	F		Overall Height	
	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Inches	Ft.	Ins.	Ft.	Ins.
10 to 15	17	0	4	8½	15	4	12	0	12	6	10	7	5
18 to 25	19	0	5	3	17	6	14	0	12	7	0	7	7
30 to 40	21	10	5	9¾	20	1	16	0	17½	7	11	7	11
45 to 65	30	0	7	1	26	10	21	0	20	12	0	9	9
70 to 100	32	0	9	0	29	9	22	0	26	12	0	9	9
125 to 150	36	6	10	0	34	6	26	0	30	14	0	9	9



Vertical Type
Refrigerating Capacity, 15 to 180 Tons

Size in Tons, Refrigerating Capacity	A		B		C		D		E	F		Overall Height	
	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Inches	Ft.	Ins.	Ft.	Ins.
15 to 25	14	6	5	0	3	10	10	0	11	5	0	6	8
25 to 35	15	6	5	0	3	10	10	0	11	5	0	6	8
40 to 50	15	0	5	0	3	6	10	0	16	5	0	7	0
			Plus 12										
60 to 100	17	6	7	0	3	6	12	0	18	5	0	7	0
			Plus 18										
120 to 180	20	0	9	6	3	6	14	0	28	5	0	7	0
			Plus 20										

TABLE 11
Worthington Pump Co. Ammonia Compressor



Steam-driven Compressors
(Layout Dimensions)

Size	A Ft.	A In.	B Ft.	B In.	C Ft.	C In.	D Ft.	D In.	E Ft.	E In.	F Ft.	F In.	G Ft.	G In.	H Ft.	H In.	J Ft.	J In.	K Ft.	K In.
8 × 6 × 9 p	11	0	2	9	3	10	3	0 *	10	0	3	4	2	0½	0	18	3	11	4	4
9 × 7 × 10 p	12	11	3	3	4	4	3	6	13	0	4	2	2	6½	0	24	4	10½	5	0½
10 × 8 × 12 p	14	3	3	7	4	7	4	0	14	0	4	6	2	6½	2	3	5	6	5	4
10½ × 8 × 12 u	15	6	3	7	4	6½	4	0	16	0	4	6	2	6½	2	3	6	3½	5	11
12 × 9 × 14 p	16	0	4	6	5	0	4	6	17	0	4	8	2	8½	2	6	6	1¾	5	9⅞
12½ × 9 × 14 u	17	9	4	0¾	4	11½	4	6	19	0	4	8	2	8½	2	6	7	2½	6	7¾
14 × 10 × 16 p	18	3	4	8	5	4	5	0	19	0	5	0	2	10	2	6	7	0	6	5¾
15 × 10 × 16 u	19	6	4	7	5	5½	5	0	20	6	5	0	3	1	2	6	8	0½	7	0¼
16½ × 11½ × 18 n	23	6	8	0	6	6	6	6	24	0	5	6	3	0	3	0	9	2	7	11
18½ × 13 × 21 n	25	6	8	9	6	6	7	0	26	0	6	0	3	0	3	3	10	2	8	6
20½ × 14 × 24 n	27	6	10	6	7	6	8	0	28	0	7	0	3	6	3	6	11	8	9	3
23½ × 16½ × 24 n	28	6	11	0	8	0	8	0	30	0	7	0	3	9	3	8	12	0	9	6

*A 36 in. × 8½ in. pulley only on 8 × 6 × 9 compressor.

p Compressors fitted with piston valve steam cylinders as illustrated.

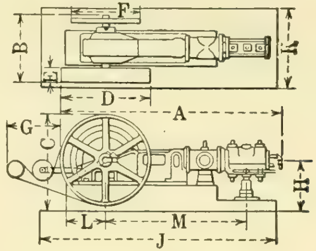
u Compressors fitted with uniflow piston valve steam cylinders (not illustrated).

n Compressors fitted with uniflow poppet valve steam cylinders (not illustrated).

Foot under cylinders not provided for 8 × 6 × 9 and 9 × 7 × 10 compressors.

TABLE 12

WORTHINGTON PUMP CO. AMMONIA COMPRESSOR



Single-belted compressors
(Layout Dimensions)

Compressor Size	A Ft.	A In.	B Ft.	B In.	C Ft.	C In.	D Ft.	D In.	E Inches	F Ft.	F In.	G Ft.	G In.	H Ft.	H In.	J Ft.	J In.	K Ft.	K In.	L Ft.	L In.	M Ft.	M In.
6 × 9	7	6	2	4¾	4	1	3	6	8½	....		2	6	2	4	10	3	3	4		18	5	5
7 ×10	10	10	3	3⅝	4	9½	4	6	8½	3	6	3	3	2	6½	11	0	4	2		24	7	5
8 ×12	11	10	3	10	5	5	5	0	10½	4	0	3	6	2	10¾	13	6	4	6	2	3	8	0¾
9 ×14	13	2	4	6⅛	6	0	5	6	12½	4	6	4	0	3	2¾	15	0	4	8	2	6	8	10
10 ×16	14	4	5	5	7	1	6	6	15½	5	0	3	9	3	9½	16	0	5	0	2	6	9	9¾

Electrically-driven Type
(Not Illustrated)

Compressor Size	A Ft.	A In.	B Ft.	B In.	C Ft.	C In.	D Ft.	D In.	E Inches	F	G	H Ft.	H In.	J Ft.	J In.	K Ft.	K In.	L Ft.	L In.	M Ft.	M In.
7 ×10	11	7	7	0	4	9	4	6	....	....	....	2	0	11	0	10	0	3	0	7	5
8 ×12	12	4	7	6	5	4	5	0	....	....	....	2	4	12	0	11	6	3	6	8	0¾
10 ×14	12	8	8	0	5	8	5	4	....	....	....	2	6	13	3	12	6	3	6	8	10
11½ ×16	15	8	9	0	6	4	6	0	....	....	....	2	10	15	6	13	0	4	0	9	9¾
13 ×18*	17	0	10	6	7	0	6	6	....	....	....	3	0	17	6	14	0	4	6	9	6
14½ ×21*	18	6	11	6	7	0	7	0	....	....	....	3	3	18	6	14	6	4	9	10	0
16½ ×24*	20	0	12	0	7	3	7	0	....	....	....	3	3	20	0	14	9	5	0	10	10

Dimensions "B" and "K" are for electrically-driven machines giving maximum width with outboard bearing motor. If a Flywheel type or Overhung motor is used, width is about the same as for a belted machine. Motor prints are necessary to secure exact dimensions.

* Do not have "Doubleseal" Stuffing Box, 18-in. stroke and over.

TABLE 13

Ingersoll-Rand Ammonia Compressor

"Imperial" Type XPV-2-A Two-stage Steam-driven Ammonia Compressors

Sea Level Operation

Duplex Steam Cylinders—Compound Steam Cylinders Furnished, Sizes Depending upon Steam Conditions

"UNIVERSAL MACHINES"

Cylinder Dimensions, Inches				Pressure Conditions, Pounds per Square Inch, Gage				Overall Dimensions, Feet and Inches			Approximate Shipping Weight, Pounds	
Diameters			Stroke	Suction—20 Discharge—185		Suction—5 Discharge—185		Length	Width	Height	Domestic	Export
Duplex steam cylinders	Ammonia cylinders: Low pressure	Ammonia cylinders: High pressure		Tons refrigeration	I. H. P. steam cylinders	Tons refrigeration	I. H. P. steam cylinders					
7*	$7\frac{1}{2}$	$5\frac{1}{4}$	10	30	48	17	39	9–0	5–1	6–4	8,000	8,800
8*	$8\frac{1}{2}$	6	12	42	66	24	54	9–10	5–4	6–5	8,700	9,600
9*	$9\frac{1}{2}$	$6\frac{3}{4}$	12	53	82	31	67	10–11	6–0	6–9	11,300	12,500
10	12	$8\frac{1}{2}$	14	88	129	51	105	12–6	7–8	7–1	19,900	22,000
12	$13\frac{1}{2}$	$9\frac{1}{2}$	16	114	166	66	135	14–0	8–9	6–8	26,000	28,500
14	15	$10\frac{1}{2}$	16	142	207	83	168	14–6	9–2	6–5	33,000	37,000
"ICE MACHINES" ONLY—20 Lb. per Square Inch Gage Suction and 185 Lb. per Square Inch Gage Discharge Pressures												
15	$16\frac{1}{2}$	$11\frac{1}{2}$	20	182	258	....	....	17–10	10–3	6–6	43,000	47,500
16	18	$12\frac{1}{2}$	20	218	310	....	....	18–5	11–0	6–11	50,300	56,000
19	23	16	24	357	496	....	....	23–8	15–6	9–6	84,000	92,500
22	26	18	30	506	702	....	....	28–4	17–0	10–0	123,000	136,000
"FREEZER MACHINES" ONLY—5 Lb. per Square Inch Gage Suction and 185 Lb. per Square Inch Discharge Pressures												
15†	$17\frac{1}{2}$	11	20	....	....	117	236	17–10	10–3	6–6	44,000	48,500
16†	20	12	20	....	....	151	303	....	....	6–11	52,000	58,000
19†	25	16	24	....	....	248	485	23–8	15–6	9–6	86,000	94,000
22†	28	18	30	....	....	348	680	28–4	17–0	10–0	125,000	137,500

* Steam Cylinders have plain piston Steam Valves—Steam Governor is Variable Speed Type.
† Low-Pressure Ammonia Cylinder has Clearance Pockets.

TABLE 14

DETAILS OF STANDARD CARBONIC COMPRESSORS

Stroke	Tons	Bore	Diameter of Piston Rod, Inches	R.p.m.	Total Horse Power	Horse Power per Ton	Suction Displacement			Flywheel		Total Weight, Pounds
							Cubic inches per minute	Cubic feet per minute	Cubic inches per min.-ton	Weight, pounds	Diameter, inches	
6	3	$2\frac{1}{4}$	$\frac{15}{16}$	120	$7\frac{1}{2}$	2.5	5,220	3.02	1740	1,000	36	2,100
6	4	$2\frac{1}{4}$	$\frac{15}{16}$	135	10	2.5	5,870	3.40	1400	1,000	36	2,100
9	5	$2\frac{1}{2}$	$1\frac{1}{8}$	95	10	2.0	7,550	4.37	1510	1,700	60	3,800
9	6	$2\frac{3}{4}$	$1\frac{1}{8}$	110	15	2.5	10,750	6.23	1800	1,700	60	3,800
10	7	$2\frac{7}{8}$	$1\frac{1}{8}$	110	20	2.8	13,200	7.65	1885	2,200	60	4,300
10	8	$2\frac{7}{8}$	$1\frac{1}{8}$	110	20	2.5	13,200	7.65	1650	2,200	60	4,300
$11\frac{1}{2}$	10	3	$1\frac{1}{4}$	90	20	2.0	13,300	7.7	1330	2,600	82	6,200
$11\frac{1}{2}$	12	$3\frac{1}{4}$	$1\frac{1}{4}$	90	25	2.07	15,900	9.22	1320	2,600	82	6,200
14	15	$3\frac{1}{2}$	$1\frac{3}{4}$	90	25	1.66	21,300	12.32	1420	3,000	82	9,500
14	18	$3\frac{5}{8}$	$1\frac{3}{4}$	85	30	1.66	21,700	12.58	1200	4,300	84	10,500
14	20	$3\frac{3}{4}$	$1\frac{3}{4}$	95	35	1.74	26,400	15.3	1320	4,500	84	11,000
14	22	$3\frac{7}{8}$	$1\frac{3}{4}$	95	40	1.8	28,300	16.38	1280	4,500	84	11,000
16	25	$4\frac{1}{4}$	2	80	40	1.6	32,300	18.69	1290	6,000	96	14,500
16	30	$4\frac{3}{8}$	2	90	50	1.66	38,700	22.4	1290	6,500	96	15,000
20	40	5	$2\frac{3}{8}$	70	60	1.5	48,700	28.21	1220	8,000	108	20,000
20	50	$5\frac{1}{8}$	$2\frac{3}{8}$	70	75	1.5	51,500	29.8	1030	9,000	108	21,000
20	60	$5\frac{3}{4}$	$2\frac{3}{8}$	75	90	1.5	66,500	38.5	1110	10,000	144	22,000
24	70	$6\frac{1}{4}$	3	65	100	1.42	88,000	50.9	1250	11,500	144	25,000
24	80	$6\frac{1}{2}$	3	65	110	1.38	96,000	55.6	1200	14,000	168	27,500
24	90	$6\frac{3}{4}$	3	65	120	1.34	101,000	58.5	1120	15,000	168	28,500
24	100	7	3	65	135	1.35	109,000	63.1	1090	16,000	168	29,500
28	120	$7\frac{3}{4}$	$3\frac{3}{4}$	60	150	1.26	120,000	69.5	1000	18,000	192	48,000
28	130	8	$3\frac{3}{4}$	60	200	1.52	128,000	74.4	990	20,000	216	50,000
28	150	$8\frac{1}{2}$	$3\frac{3}{4}$	60	275	1.82	147,000	85.2	970	25,000	216	55,000

CHAPTER III

THE ABSORPTION REFRIGERATING MACHINE

History and Development.—The absorption machine was one of the first forms of refrigerating devices. It is based on an entirely different principle from that of the compression machine, namely that of driving a gas out of solution by the direct application of heat and (after evaporation in the refrigerating coils) of absorbing this gas into solution again in order to complete the cycle. The first mention of such a method is that by Professor Leslie who, about 1810, described a laboratory method of making ice by freezing water in a saucer placed in a receiver under a vacuum—the water vapor as it boiled off under the low pressure maintained in the receiver being absorbed by sulphuric acid in another saucer. Much later, in 1878, Windhauser made a complete cycle, using heat to drive the water out of solution and to obtain a concentrated solution again. Edmund Carré[1] in 1850 developed a small sulphuric acid refrigerating machine of the intermittent type, and the ammonia absorption machine of the present day is the old Carré perfected.

The Modern Design.—In the ammonia absorption machine, the *generator* (Fig. 55) contains a concentrated solution of ammonia and water and heat is applied[2] by means of steam coils submerged in the solution. The gas driven off is later condensed in an ordinary ammonia condenser and on evaporation in the refrigerating coils is reabsorbed in the *absorber* (Figs. 56 and 57). The absorption medium is the weaker aqueous solution of ammonia from the generator, which has to be cooled to some temperature in the neighborhood of 100 deg. F. or lower in order to work with concentrations required in the present practice. In 1870 Mart devised the *economizer* or *exchanger* (a modern design of which is shown in Fig. 58) whereby this weaker solution was cooled by a counter current flow exchange of heat with the stronger and colder solution from the absorber. This operation permitted the stronger

[1] Described by Dr. John Hopkinson in 1882 before the Society of Arts. Such an ice plant made about 12 tons of ice per 24 hours in 650-lb. blocks.

[2] In the household refrigerating machine, gas or electric resistance coils provide the necessary heating.

solution to return to the generator at as high a temperature and the weaker solution to arrive at the absorber at as low a temperature as possible. The absorption machine devised by Pontifex and Wood in 1876 included the *analyzer* containing trays invented by Reece to separate from the vapors leaving the generator some of the 5 to 15 per cent of water vapor which is present in the ammonia vapor as well as the particles of liquid (priming) which are carried on by the action of boiling in the generator. The analyzer is really a part of the generator, but it is omitted in some designs. At present a *rectifier* (or dehydrator) is also used in order to condense out as much of the water vapor as can safely be attempted by cooling these gases in a sort of water-cooled pre-cooler or preliminary condenser where only part of the superheat in the ammonia is taken away. A liquid pump, the only moving part of the

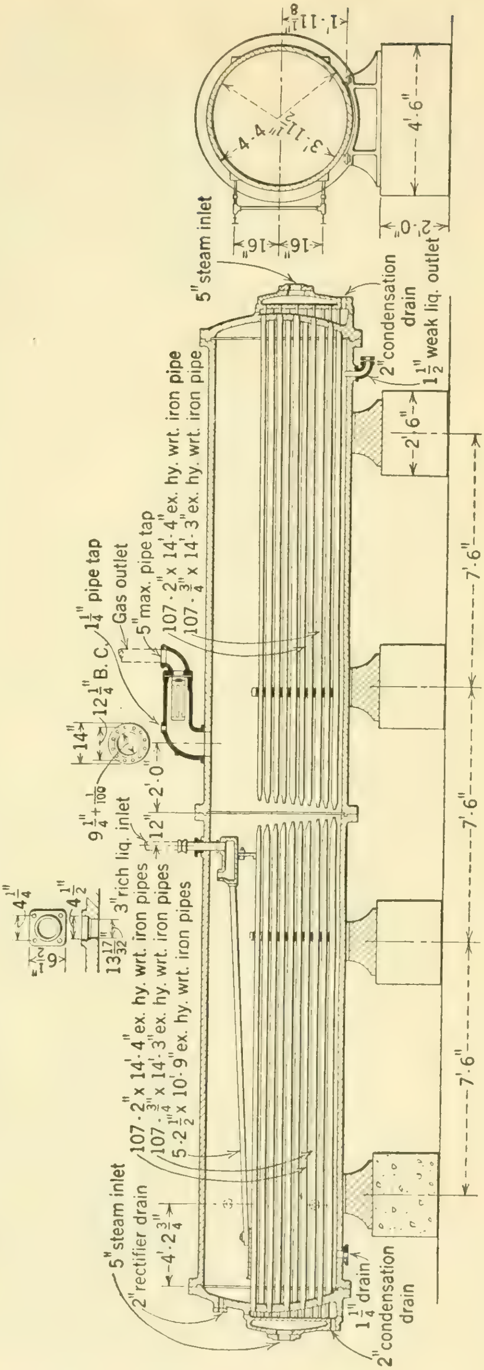


Fig. 55.—The Generator.

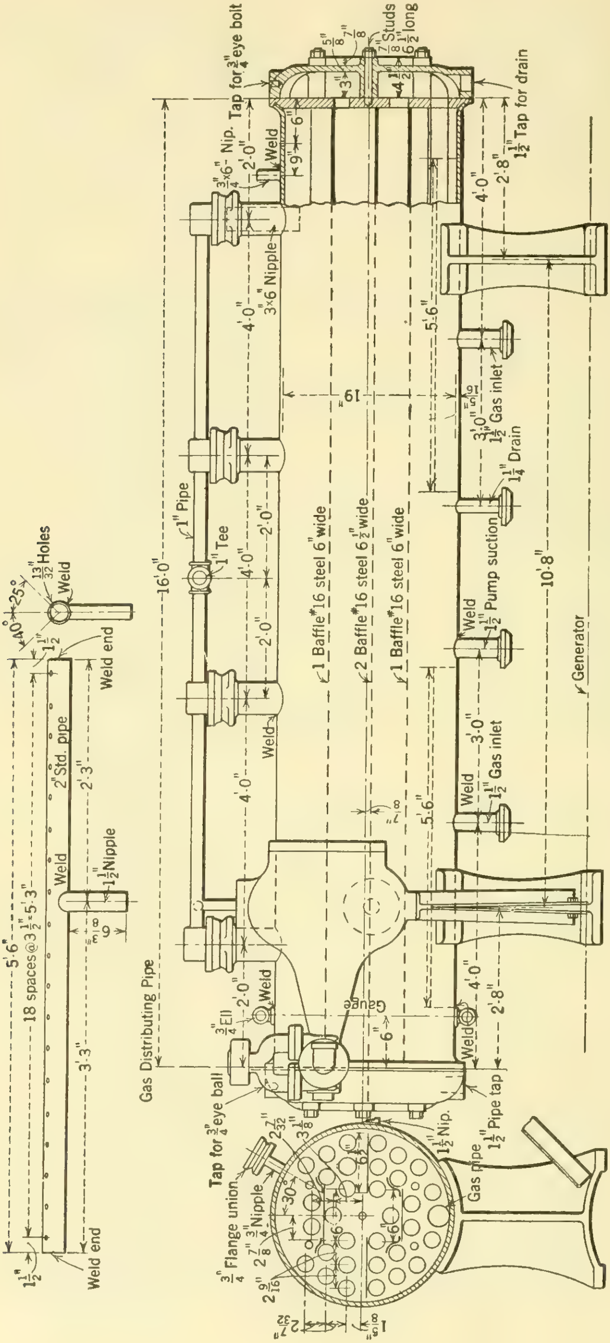


Fig. 56.—The Absorber.

machine, returns the strong aqua to the generator, the complete cycle being shown in Fig. 59.

The absorption machine has never been very popular in the United States for general refrigeration (but much more so than in Great Britain) although in special work it has certain advantages over the compression type: for example, those plants having a large amount of exhaust steam for which no other use can be found frequently can use the absorption in preference to the compression machine. This appears to be the case in oil refineries and (to some extent) in packing plants, where usually exhaust steam is plentifully available. The use of high-pressure steam in the coils of the generator is now no longer advocated although many such plants were installed up to about 1915. The use of the absorption machine for the production of low temperatures is now no longer necessary, although for a long time low temperature work was considered the field for the absorption machine only. In this respect it should be noted that the absorber liquefies and then absorbs the gas from the evaporating coils and is in no manner affected by the pressure exerted by the gas (and in consequence the specific volume of this gas) provided the pipe from the evaporator to the absorber is large enough to accommodate the increased volume at these low pressures, and that the absorber can spray the entering gas properly with the weak solution. The specific volume of the ammonia therefore is not an important factor in the absorption machine, though it is extremely important in the compression type, both because of the piston displacement required and also because of the decreased volumetric efficiency occasioned by the cylinder walls and ports. As stage ammonia compression has been developed so that it can take low temperature refrigeration with economy the use of the absorption machine for low temperature work is no longer necessary, with the result that, as mentioned, the best field for this machine is where exhaust steam would otherwise be thrown away.

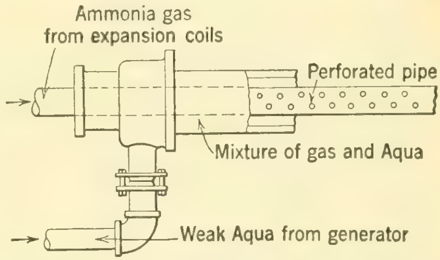


FIG. 57.—Weak Aqua and Gas Mixer.

The principle of the operation of the absorption machine[3] is the ability of water to absorb large quantities of ammonia—as much at

[3] The values of the p, t, x relations of aqua ammonia are given by Thomas Wilson in Bulletin No. 146 of the Engineering Experiment Station of the University of

times as 1000 volumes—depending in amount on the temperature and the pressure of the solution (Table 18). For example a concentration of 42.5 per cent is obtained in the absorber under a pressure of 35 lb. abs. (20 lb. gage) and 80 deg. F., whereas at a temperature of 200 degrees and a pressure of 165 lb. abs. (150 lb. gage) the maximum concentration that can be held in solution is 35.5 per cent, and the difference, 7.0 per cent, will represent the amount to be boiled out of solution in the generator.

Action of the Generator.—The action in the generator is not as clearly understood as it might be, but the following explanation is at

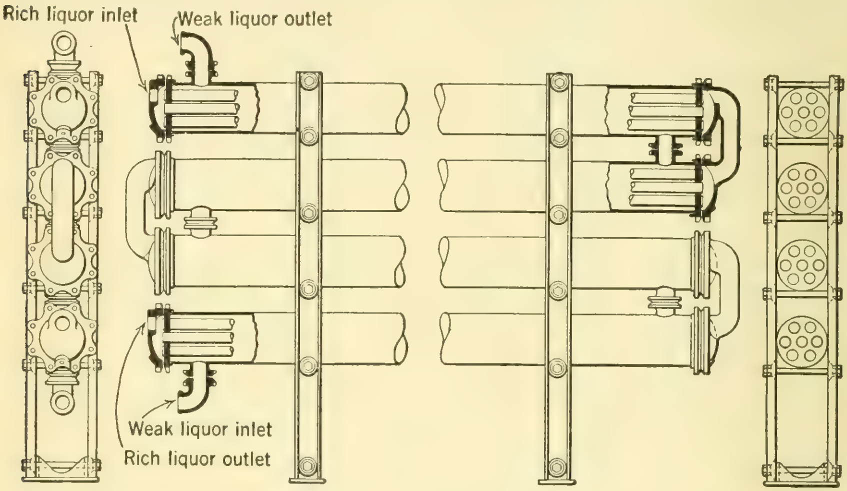


FIG. 58.—The Exchanger.

least reasonable. The heat supplied by the steam in the steam coils can be considered to cause the following action:

1. To break up the bond of association between the ammonia and the water.

2. To cool the resulting anhydrous ammonia to the temperature corresponding to saturation for the pressure at hand.

Illinois. The equation connecting these variables is given by G. A. Goodenough as,

$$\frac{\theta}{T} = \frac{1}{1 + 0.70356(1 - z^2)}$$

$$z = \sqrt{x + 0.05}(1.347 - 2.9x + 1.77x^2)$$

θ = the temperature of saturated ammonia vapor, deg. abs.
T = the temperature of ammonia solution, deg. abs
x = mol concentration of ammonia.

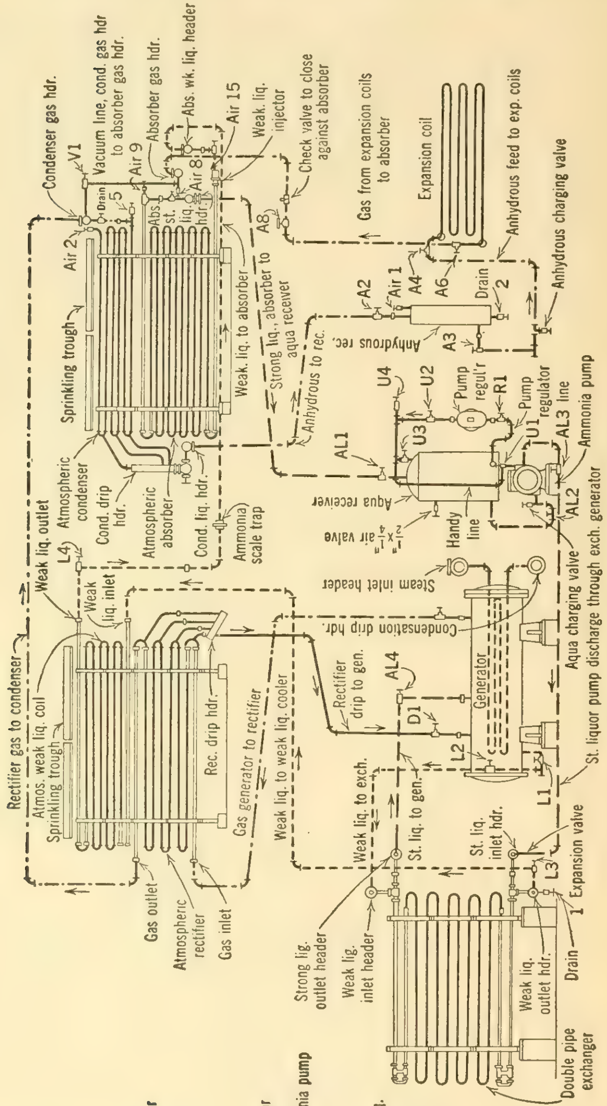
TABLE 15

Heat Removed in Absorber for Different Values of Strong and Weak Aqua

Per Cent of Weak Aqua	Temperature and Concentration of Strong Aqua																																			
	25 per cent									30 per cent									35 per cent									40 per cent								
	60 deg. F.			80 deg. F.			100 deg. F.			60 deg. F.			80 deg. F.			100 deg. F.			60 deg. F.			80 deg. F.			100 deg. F.			60 deg. F.			80 deg. F.			100 deg. F.		
	Temperature Weak Aqua									Temperature Weak Aqua									Temperature Weak Aqua									Temperature Weak Aqua								
	80	100	120	100	120	140	120	140	160	80	100	120	100	120	140	120	140	160	80	100	120	100	120	140	120	140	160	80	100	120	100	120	140	120	140	160
10 G	6.0	6.0	6.0	6.0	6.0	6.0	6.0	6.0	6.0																											
H	853	954	1056	838	939	1041	821	924	1026																											
15 G	$8\frac{1}{2}$	$8\frac{1}{2}$	$8\frac{1}{2}$	$8\frac{1}{2}$	$8\frac{1}{2}$	$8\frac{1}{2}$	$8\frac{1}{2}$	$8\frac{1}{2}$	$8\frac{1}{2}$	$5\frac{2}{3}$	$5\frac{2}{3}$	$5\frac{2}{3}$	$5\frac{2}{3}$	$5\frac{2}{3}$	$5\frac{2}{3}$	$5\frac{2}{3}$	$5\frac{2}{3}$	$5\frac{2}{3}$																		
H	892	1046	1200	877	1032	1186	862	1016	1173	828	923	1019	812	908	1004	795	891	989																		
20 G	16	16	16	16	16	16	16	16	16	8	8	8	8	8	8	8	8	8	$5\frac{1}{3}$	$5\frac{1}{3}$	$5\frac{1}{3}$	$5\frac{1}{3}$	$5\frac{1}{3}$	$5\frac{1}{3}$	$5\frac{1}{3}$	$5\frac{1}{3}$	$5\frac{1}{3}$									
H	1035	1344	1656	1021	1332	1644	1006	1318	1636	862	1007	1153	848	993	1139	831	976	1125	799	889	979	783	873	963	766	856	948									
25 G	....	....	....	....	....	....	....	....	....	15	15	15	15	15	15	15	15	15	$7\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{1}{2}$	5	5	5	5	5	5	5	5	5
H	....	....	....	....	....	....	....	....	....	995	1285	1579	980	1274	1568	964	1258	1559	830	966	1102	814	950	1087	793	934	1074	769	852	936	752	836	920	735	819	905
30 G	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	14	14	14	14	14	14	14	14	14	7	7	7	7	7	7	7	7	7
H	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	952	1224	1499	937	1211	1487	921	1197	1480	796	922	1047	779	906	1033	762	890	1020
35 G	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	13	13	13	13	13	13	13	13	13
H	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	907	1160	1416	891	1147	1404	876	1132	1397

G = lb. of Strong Aqua per 1 lb. of Ammonia absorbed.

H = B.t.u. absorbed by the cooling water per 1 lb. of Ammonia entering solution.



Valves

A1 - Gas valve Gen. to rec.
A2 - Anhydrous-condenser to rec.
A3 - Anhydrous feed from rec.
A4 - Expansion valves at coils
A6 - Gas valves, coil outlets
A8 - Gas expansion coils to absorber
L1 and L2 - Wk. liq. valves at generator
L3 - Exp. V. exchanger to wk. liq. coil
L4 - Wk. liq. from W.L. coil to abs.
AL1 - St. liq. absorber to aqua rec.
AL2 - Pump suction valve
AL3 - Pump discharge valve
AL4 - Pump discharge valve at generator
D1 - Rectifier drip valve to generator
U1 - Valve on discharge chamber ammonia pump
U2 - Valve above pump regulator
U3 - Valve above aqua receiver
U4 - Valve to air from handy line
R1 - Valve on pump req. line bottom req.
Drain 1 - Valve at bottom of exch.
Drain 2 - Valve at bottom anhyd. rec.
Drain 5 - Valve at bottom of condenser
V1 - Valve on vacuum line, between cond. and abs. gas hdrs.
Air 1 - Valve at top anhyd. rec.
Air 2 - Valve at top of cond.
Air 8 - At bottom of vacuum line
Air 9 - At top of absorber coils
Air 15 - At bottom of abs. coils

FIG. 59.—The Absorption Machine Complete.

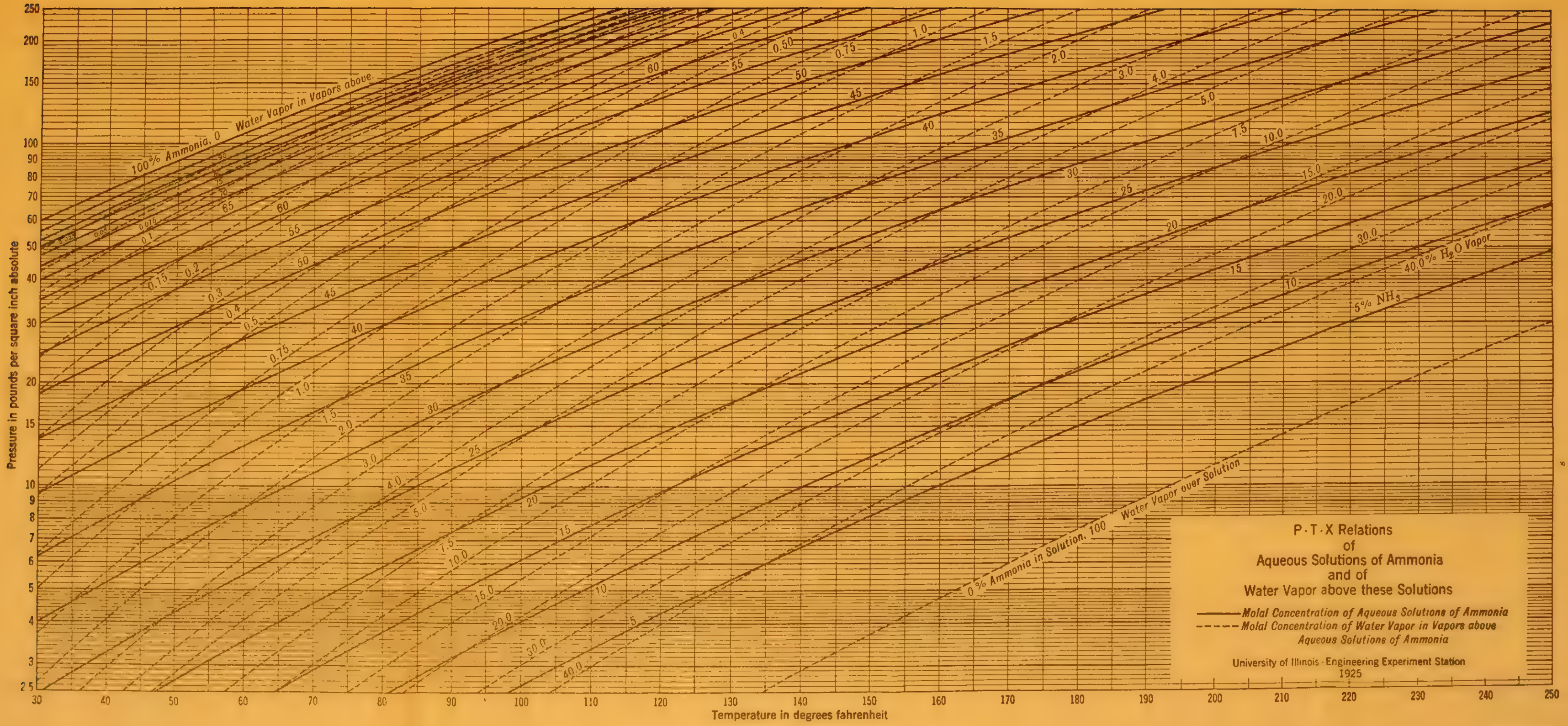


Fig. 60.—Chart of Properties of Aqueous Solutions of Ammonia.

To face page 89

3. To vaporize the liquid ammonia.

4. To heat the entering strong aqua from the absorber and the exchanger to the temperature of the generator, and, finally,

5. To evaporate a certain amount of water vapor which passes off into the rectifier with the vapor boiled off. The heat of solution is given by the formula [4]

$$q = 345(1 - x) - 400x^2$$

where x is the average concentration of the solution. The following problem will bring out the method of calculation.

Problem.—A strong aqueous solution enters the generator at 160 deg. F. and a concentration of 35 per cent, and the temperature of the solution in the generator is 215 degrees. The weak aqua leaving the generator has a concentration of 30 per cent and the pressure in the generator is 145 lb. per square inch. Water vapor to the amount of 7.4 per cent is boiled off with the ammonia.

The amount of strong aqua necessary to be circulated in order to boil off one pound of ammonia is given by the relation,

$$nx_1 - (n - 1)x_2 = 1$$

where x_1 = the concentration of the strong
x_2 = the concentration of the weak

and therefore

$$n = \frac{1 - x_2}{x_1 - x_2} = \frac{1.0 - 0.30}{0.35 - 0.30} = 14.0 \text{ lb.}$$

The heat supplied by the steam coils in the generator then becomes:

a. Heat of disassociation

$q = 345(1 - x) - 400x^2 = 345 \times 0.675 - 400 \times 0.325^2 = 190.8$ B.t.u. per 1.0 lb. ammonia boiled out.

b. Heat necessary to cool the liquid ammonia from 215 to 76.8 deg.

$= -305 + 128.4 = -176.6$ B.t.u.

c. Heat necessary to vaporize the ammonia (at 145 lb.)

$= 501.9$ B.t.u.

d. Heat necessary to superheat this gas to 215 degrees

$= 88.3$ B.t.u.

e. Heat required to raise the temperature of the entering strong aqua[5]

$= 1.132 \times (215 - 160) \times 14.0 = 871$ B.t.u.

f. Finally the heat necessary to boil off 0.074 lb. of water vapor at 215 deg. F. is

$= 0.074 \times 969.9 = 71.8$ B.t.u.

[4] H. Mollier, V. D. I., 1907. Goodenough, Principles of Thermodynamics.
[5] Ammonia will begin to boil off at 192 deg. F.

TABLE 16

Table of Capacities and Surfaces in Tons of Refrigeration

Square feet of surface in Generator, Absorber, and Cooling Water per ton of Refrigeration under varying temperatures and pressures. Also Standard and Minimum steam pressure on Generator and factors for determining required ice making surfaces of Con., W. L. Coolers, Exch. and Rect. from given refrigerating capacity.

Brine Temperature		Cooling Water 55 Deg. F.									Cooling Water 60 Deg. F.									Tons R / Factor = tons of ice		
		Generator			Absorber, square feet per ton						Generator			Absorber, square feet per ton								
		Square feet per ton	Steam pressure	Pounds of steam per hour per ton	Horizontal tubular	Multipipe	Atmospheric	Dry only	D. P.	Gallons of water per minute per ton	Square feet per ton	Steam pressure	Pounds of steam per hour per ton	Horizontal tubular	Multipipe	Atmospheric	Dry only	D. P.	Gallons of water per minute per ton	Cond. and Ammonia pump	W. L. Cooler	Exch. and Rect.
15	Std.	14.5	3.0	29	22	14	18	16	6.0	2.00	14.0	5.0	30	23	15	19	17	7.0	2.25			
0	Min.	15.5	1.0	30	23	15	19	17	6.5	2.25	15.0	3.0	31	24	16	20	18	7.5	2.50			
10	Std.	14.5	3.0	29	22	14	18	16	6.0	2.00	14.0	5.0	30	23	15	19	17	7.0	2.25			
0	Min.	15.5	1.5	30	23	15	19	17	6.5	2.25	15.0	3.0	31	24	16	20	18	7.5	2.50			
5	Std.	14.5	3.0	30	22	14	18	16	6.0	2.00	14.0	5.0	31	23	15	19	17	7.0	2.25			
0	Min.	15.5	2.0	31	23	15	19	17	6.5	2.25	15.0	3.0	32	24	16	20	18	7.5	2.50	1.7	1.75	1.75
0	Std.	14.5	3.5	30	23	15	19	17	6.5	2.25	14.0	5.0	31	24	16	20	18	7.5	2.25			
	Min.	15.5	2.5	31	24	16	20	18	7.0	2.50	15.0	3.5	32	25	17	21	19	8.0	2.50			
0	Std.	15.0	4.5	31	23	15	19	17	6.5	2.25	14.5	6.0	32	24	16	20	18	7.5	2.75			
5	Min.	16.0	3.0	32	24	16	20	18	7.0	2.50	15.5	4.5	33	25	17	21	19	8.0	2.50			
0	Std.	15.0	5.5	31	23	15	19	17	6.5	2.25	14.5	7.0	32	24	16	20	18	7.5	2.75			
10	Min.	16.0	3.5	32	24	16	20	18	7.0	2.50	15.5	5.0	33	25	17	21	19	8.0	2.75			
0	Std.	15.0	6.0	32	24	16	20	18	7.0	2.50	14.5	8.0	33	25	17	21	19	8.0	3.00			
5	Min.	16.0	4.0	33	25	17	21	19	7.5	2.75	15.5	6.0	34	26	18	22	20	8.5	2.75			
0	Std.	15.0	7.5	33	6*	..	..	19	....	2.50	14.5	10.0	34	6*	..	..	20	....	3.00			
20	Min.	16.0	5.0	34	7*	..	..	20	....	2.75	15.5	7.5	35	7*	..	..	21					

Water run in series Fig. Abs. 10 deg. higher. Water run in series Fig. 10 deg. higher on abs.

Brine Temperature		Cooling Water 65 Deg. F.									Cooling Water 70 Deg. F.									Tons R / Factor = tons of ice		
		Generator			Absorber, square feet per ton						Generator			Absorber, square feet per ton								
		Square feet per ton	Steam pressure	Pounds of steam per hour per ton	Horizontal tubular	Multipipe	Atmospheric	Dry only	D. P.	Gallons of water per minute per ton	Square feet per ton	Steam pressure	Pounds of steam per hour per ton	Horizontal tubular	Multipipe	Atmospheric	Dry only	D. P.	Gallons of water per minute per ton	Cond. and Ammonia pump	W. L. Cooler	Exch. and Rect.
15	Std.	13.5	7	30	24	16½	20	18.5	8.0	2.75	13.0	9	31	26	18	21	20.5	9.5	3.25			
0	Min.	14.5	4	31	25	17½	21	19.5	8.5	3.00	14.0	6	32	27	19	22	21.5	10.0	3.50			
10	Std.	13.5	7	30	24	16½	20	18.5	8.0	2.75	13.0	9	31	26	18	21	20.5	9.5	3.25			
0	Min.	14.5	4	31	25	17½	21	19.5	8.5	3.00	14.0	6	32	27	19	22	21.5	10.0	3.50			
5	Std.	13.5	8	30	24	16½	20	18.5	8.0	2.75	13.0	10	31	26	18	22	20.5	9.5	3.25			
0	Min.	14.5	5	31	25	17½	21	19.5	8.5	3.00	14.0	7	32	27	19	23	21.5	10.0	3.50	1.75	1.8	1.8
0	Std.	13.5	8	31	25	17	21	19.0	8.5	3.00	13.0	10	32	27	19	22	21.0	10.0	3.50			
	Min.	14.5	5	32	26	18	22	20.0	9.0	3.25	14.0	7	33	28	20	23	22.0	10.5	3.75			
0	Std.	14.0	9	31	25	17	21	19.0	8.5	3.00	13.0	12	32	27	19	22	21.0	10.0	3.50			
5	Min.	15.0	6	32	26	18	22	20.0	9.0	3.25	14.5	9	33	28	20	23	22.0	10.5	3.75			
0	Std.	14.0	10	32	25	18	22	19.0	9.0	3.25	13.5	13	33	27	20	23	22.0	10.5	3.50			
10	Min.	15.0	7	33	26	19	23	20.0	9.5	3.50	14.5	10	34	28	21	24	23.0	11.0	3.75			
0	Std.	14.0	12	32	26	18	22	20.0	9.0	3.25	13.5	14	33	6*	20	23	22.0	11.0	3.75			
15	Min.	15.0	8	33	27	19	23	21.0	9.5	3.50	14.5	11	34	7*	21	24	23.0	11.5	4.00			
0	Std.	14.0	14	33	6*	...	..	.....	....	3.50	13.5	15	34	6*	...	..	.....	....	3.75			
20	Min	15.0	10	34	7*	...	..	.....	....	3.75	14.5	12	35	7*	...	..	.....	....	4.00			

* Use Dry Abs. of surf. indicated in connection with Hor. Tub. Surf. for −15 deg. brine.

TABLE 16—*Continued*

Always to be used for standard ice making plants.

Brine Temperature		Cooling Water 75 Deg. F.									Cooling Water 80 Deg. F.									Tons R / Factor = tons of ice		
		Generator			Absorber, square feet per ton						Generator			Absorber, square feet per ton								
		Square feet per ton	Steam pressure	Pounds of steam per hour per ton	Horizontal tubular	Multipipe	Atmospheric	Dry only	D. P.	Gallons of water per minute per ton	Square feet per ton	Steam pressure	Pounds of steam per hour per ton	Horizontal tubular	Multipipe	Atmospheric	Dry only	D. P.	Gallons of water per minute per ton	Cond. and Ammonia pump	W. L. Cooler	Exch. and Rect.
15	Std.	12.5	10	31	28	20	22	23	11.0	3.75	13.0	15	32	31	22	24	26	13.0	4.75			
0	Min.	13.5	8	32	29	21	23	24	11.5	4.00	14.0	13	33	32	23	25	27	13.5	5.00			
10	Std.	12.5	10	31	28	20	22	23	11.0	3.75	13.0	15	32	31	22	24	26	13.0	4.75			
0	Min.	13.5	8	32	29	21	23	24	11.5	4.00	14.0	13	33	32	23	25	27	13.5	5.00			
5	Std.	12.5	10	32	28	20	23	23	11.0	4.00	13.0	15	33	32	22	24	26	13.5	5.00	1.82	1.85	1.85
0	Min.	13.5	8	33	29	21	24	24	11.5	4.25	14.0	13	34	33	23	25	27	14.0	5.25			
0	Std.	13.0	12	32	29	21	23	24	11.5	4.00	13.0	18	33	32	23	25	27	13.5	5.00			
	Min.	14.0	9	33	30	22	24	25	12.0	4.25	14.0	15	34	33	24	26	28	14.0	5.25			
0	Std.	13.0	12	32	29	21	24	24	11.5	4.25	13.5	20	33	33	23	25	27	14.0	5.25			
5	Min.	14.0	10	33	30	22	25	25	12.0	4.50	14.5	16	34	31	24	26	28	14.5	5.50			
0	Std.	13.0	14	33	30	22	24	25	12.0	4.25	13.5	21	34	33	24	26	28	14.5	5.50			
10	Min.	14.0	12	34	31	23	25	26	12.5	4.50	14.5	17	35	34	25	27	29	15.0	5.75			
0	Std.	13.5	15	34	6*	23	25	25	12.5	4.50	14.0	22	35	6*	...	...	...	...	5.75			
15	Min.	14.5	13	35	7*	24	26	26	13 0	4.75	15.0	18	36	7*	...	...	...	...	6.00			
0	Std.	13.5	17	35	6*	...	...	...	...	4.75	14.0	24	36	6*	...	...	...	...	6.00			
20	Min.	14.5	15	36	7*	...	...	...	...	5.00	15.0	20	37	7*	...	...	...	...	6.25			

Water run in series Fig. Abs. 10 deg. higher. Water 80 deg. to Absorber.

Brine Temperature		Cooling Water 85 Deg. F.									Cooling Water 90 Deg. F.									Tons R / Factor = tons of ice		
		Generator			Absorber, square feet per ton						Generator			Absorber, square feet per ton								
		Square feet per ton	Steam pressure	Pounds of steam per hour per ton	Horizontal tubular	Multipipe	Atmospheric	Dry only	D. P.	Gallons of water per minute per ton	Square feet per ton	Steam pressure	Pounds of steam per hour per ton	Horizontal tubular	Multipipe	Atmospheric	Dry only	D. P.	Gallons of water per minute per ton	Cond. and Ammonia pump	W. L. Cooler	Exch. and Rect.
15	Std.	13.0	20	33	34	25	26	30	15	6.0	13.5	28	34	39	29	30	35	18	7.5			
0	Min.	14.0	18	34	35	26	27	31	16	6.5	14.5	25	35	40	40	31	36	19	8.0			
10	Std.	13.0	20	33	34	25	26	30	15	6.0	13.5	28	34	39	29	30	35	18	7.5			
0	Min.	14.0	18	34	35	26	27	31	16	6.5	14.5	24	35	40	30	31	36	19	8.0			
5	Std.	13.5	21	34	34	26	26	31	15	6.5	13.5	30	35	39	30	31	36	18	8.0			
0	Min.	14.5	18	35	35	27	27	32	16	7.0	14.5	26	36	40	31	32	37	19	8.5			
0	Std	13.5	23	35	35	26	27	31	16	6.5	14.0	32	36	40	30	31	36	19	8.0	1.9	1.9	1.9
	Min.	14.5	20	36	36	27	28	32	17	7.0	15.0	28	37	41	31	32	37	20	8.5			
0	Std.	14.0	25	36	35	27	27	32	16	7.0	14.0	34	37	40	31	32	37	19	8.5			
5	Min.	15.0	22	37	36	28	28	33	17	7.5	15.0	30	38	41	32	33	38	20	9.0			
0	Std.	14.0	30	37	36	27	28	32	17	7.0	14.5	40	38	41	33	33	37	20	8.5			
10	Min.	15.0	26	38	37	28	29	33	18	7.5	15.5	36	39	42	34	34	38	21	9.0			

Water 85 deg. to Absorber. Water 90 deg. to Absorber.

* Use Dry Abs. of surf. indicated in connection with Hor. Tub. surf. for − 10 deg. brine.

TABLE 16—*Continued*

Temperature of Water, Degrees		55	60	65	70	75	80	85	90
Type		Surface, Square Feet per Ton							
Condensers:									
Good water ①									
Double pipe	Ice	15.0	16.5	18.0	19.5	21.0	23.0	25.0	28.0
	Ref.	8.8	9.7	10.3	11.3	11.5	12.6	13.2	14.7
Bad water ①									
Atmospheric	Ice	.....	44.0	47.0	50.0	53.0	57.0	61.0	65.0
	Ref.	.....	26.0	27.0	28.2	29.6	31.2	33.0	35.0
Good water ②									
Horizontal tabular	Ice	30.0	34.0	38.0	42.0	46.0	50.0	54.0	60.0
Where have less room	Ref.	17.6	20.0	21.7	24.0	25.2	27.5	29.4	31.6
Good water ③									
Multitube	Ice	17.0	18.5	20.0	21.5	23.0	25.0	27.0	30.0
	Ref.	10.0	10.9	11.4	12.3	12.6	13.8	14.2	15.8
Weak liquor cooler:									
Good water ①									
Double pipe	Ice	5.0	5.5	6.0	6.5	7.0	7.5	8.3	9.3
	Ref.	2.9	3.3	3.5	3.8	4.1	4.4	4.9	
Bad water ①									
Atmospheric	Ice	.....	11.0	11.75	12.50	13.25	14.25	15.25	16.25
	Ref.	.....	6.5	6.9	7.35	7.8	8.4	9.0	9.6
Good water ②									
Multitube	Ice	5.66	6.16	6.66	7.16	7.66	8.33	9.0	10.0
Where less space	Ref.	3.33	3.62	3.80	4.1	4.2	4.58	4.74	5.26
Distilled water cooler:									
Double pipe	Ice	5.5	6.0	6.4	6.8	7.2	7.6	8.0	8.5

Horizontal multipass brine cooler	Brine Temperature, Degrees F. Range = Approximately 10 Deg.					NOTE For single pass brine coolers for ice making tanks, use 35 sq. ft. per ton ice.
	+10	0	−5	−10	−15	
	Surface per Ton Refrigeration					
	11	12	14	15	16	
Exchanger	Double pipe = 9.2 sq. ft. per ton ice. Double pipe = 5.3 sq. ft. per ton ref.					Takes up more room. Use on units of 25 tons ice or less.
	Multitube = 10.56 sq. ft. per ton ice. Multitube = 5.86 sq. ft. per ton ref.					Takes up less room. Use on units over 25 tons ice.
Rectifier	Double pipe = 4.86 sq. ft. per ton ice. Double pipe = 2.85 sq. ft. per ton ref.					Same as for Exchanger.
	Multitube = 6.6 sq. ft. per ton ice. Multitube = 3.9 sq. ft. per ton ref.					Same as for Exchanger.

TABLE 16—*Continued*

AMMONIA PUMPS

Capacity, Tons		Size	Capacity in Pounds at 28 Deg. B. Aqua Ammonia per Minute, 36 r.p.m.	Floor Space, Inches
Ice	Ref.			
		St. Cyl. × NH_3 Cyl. × Stroke		
7	13	6 × $2\frac{3}{4}$ × 8	26.2	12×56
10	18	8 × $3\frac{1}{4}$ × 8	63.4	12×56
12	21	8 × $3\frac{1}{2}$ × 8	75.1	12×56
15	26	8 × $3\frac{1}{4}$ × 12	92.2	19×80
18	31	8 × $3\frac{1}{2}$ × 12	108.7	19×80
25	46	10 × $4\frac{1}{4}$ × 12	163.4	19×81
30	52	10 × $4\frac{1}{2}$ × 12	184.9	19×81
40	73	12 × $5\frac{1}{4}$ × 12	256.6	20×81
45	80	12 × $5\frac{1}{2}$ × 12	283.0	20×81
50	96	14 × 6 × 12	339.5	21×83
60	114	14 × $6\frac{1}{2}$ × 12	400.8	21×83
100	200	16 × $7\frac{1}{2}$ × 16	705.2	27×97
130	228	16 × 8 × 16	806.4	27×97
130	228	18 × 8 × 16	806.4	29×97

The total heat absorbed, per 1.0 lb. of ammonia boiled off, = 1547.2 B.t.u. In the rectifier the water content will be reduced to 0.2 per cent and the difference, 7.2 per cent, will be condensed and will form a strong aqua of 68 per cent concentration. There will be, then, a return to the generator of 15.3 per cent of ammonia, and the value 1547.2 [neglecting the action of the strong aqua] must be increased to $\frac{1547.2}{0.847} = 1827$ B.t.u. Under standard conditions of operation (5 deg. F. and 145 lb. abs. in this case) 0.412 lb. of ammonia per minute are required or $0.412 \times 1827 = 753$ B.t.u. If each pound of steam supplies 964 B.t.u. then $\frac{753 \times 60}{964} = 46.9$ lb. of steam at 6.0 lb. gage pressure would be required per ton of refrigeration per hour, an amount somewhat greater than that given in the tables of capacities and surfaces and used in the specification of absorption machines. It will be noted that the value 46.9 can be reduced by using a greater difference in the concentrations either by means of a higher steam pressure or a colder cooling water in the absorber. The fact that water boils as well as ammonia causes some operating difficulties and necessitates the use of a rectifier which functions as a precooler but is carefully operated so as to remove part and not all of the superheat, thereby reducing the temperature of the gas to a point where the greater part of the water vapor will condense out. This action as well as the action of the entire cycle is well described in Bulletin No. 146 of the Engineering Experiment Station, University of Illinois, a part of which is abstracted in the following.

APPLICATION TO AMMONIA ABSORPTION PROCESS

[From Bulletin No. 146, University of Illinois Experiment Station]

Problem.—The use of the tables and diagrams can be brought out best by a study of the operation of an ammonia absorption refrigerating machine. In order to avoid indefinite generalities a particular case will be taken.

In this problem the ammonia solution enters the generator with a concentration of 35 per cent, and leaves after the boiling off of the ammonia with a reduced concentration of 25 per cent. The condenser pressure is 150 lb. per square inch absolute, which will be taken as the generator pressure likewise. The average concentration in the generator may be considered as 30 per cent. The given generator pressure will be reached when the solution has been heated to 217.4 deg. F. (Fig. 60) by the steam admitted to the heating coils. Under these conditions the vapors leaving the generator will show a water content of 7.4 per cent, corresponding to a partial pressure of 11.10 lb. for the water vapor and 138.9 lb. for the ammonia.

On passing through the rectifier the vapors undergo partial condensation by cooling. The usual practice is to cool the vapors to a temperature approximately 20 degrees above the condensation point of anhydrous ammonia at the pressure in use. In the present case the vapors will be cooled to a temperature of 98.81 degrees as will be seen by reference to the tables of the Bureau of Standards.[8] This corresponds to a concentration of ammonia in the solution phase of 71.3 per cent, and (for a temperature of 98.8 deg. F. and a concentration of ammonia of 71.3 per cent) a concentration of water in the vapor phase of 0.160 per cent. In other words, in their passage through the rectifier the vapors have been robbed of part of their contained water, the partial pressure of which has been reduced to 0.25 lb., while that of the ammonia has been increased to 149.75 lb.

To do this some of the ammonia vapor evolved in the generator has been sacrificed. It is impossible to return any of the original water in the vapor to the generator without returning such an amount of ammonia that the concentration of the solution formed will be in equilibrium with the vapors remaining after the condensation. Likewise it is impossible to cause this condensation without a cooling of the solution, unless the vapor pressure be materially increased.

Solution of Problem.—Consideration of the case of 2000 pounds of vapor mixture in the rectifier after leaving the generator will show the extent of this process:

Water content of vapor from generator	7.4	per cent
Weight of water	148.0	lb.
Weight of ammonia	1852.0	lb.
Ammonia content of solution returned to generator from rectifier	71.3	per cent
Weight of water returned	144.8	lb.
Weight of ammonia	359.7	lb.
Total	504.5	lb.
Water content of gases sent on to receivers	0.160	per cent
Weight of water passed	2.393	lb.
Weight of ammonia	1493.10	lb.
Total	1495.50	lb.
Partial pressure of water vapor	0.25	lb. per square inch
Partial pressure of ammonia	149.75	lb. per square inch

[8] B. S. Circular No. 142, 1923.

On flowing through the pressure reducing, or so-called expansion, valve the ammonia solution experiences a release in pressure to a value determined by the concentration of the aqua ammonia in the absorber and by the temperature of the latter. Since the pressure will be greater the higher the concentration, and since no concentration changes occur during the pumping of the solution from the absorber to the generator, it follows that the back pressure in the cooling coils will be determined in the present instance by the vapor pressure of a 35 per cent ammonia solution at whatever temperature the absorber may be kept during the operation of the machine.

It is desired to maintain a temperature of 6 deg. F. in the cooling coils. Although further experimental work concerning these lower temperature vapor pressures must be done before the tables can be extended into this region, a fairly satisfactory extrapolation can be performed by the use of the aqueous vapor content table. There it will be noticed that as a general rule an increase in temperature of approximately 41.0 deg. F. will cause an increase of about 5 per cent in the composition of the solution with which a given composition of the vapor phase is in equilibrium. This rule may be applied in order to calculate from the data furnished in the table for a temperature of 32 deg. F. the solution with which a given vapor phase will be in equilibrium at 6 deg. F. The quantity calculated in this manner as the amount to be either subtracted from or added to the values there tabulated may be taken as 3.17 per cent.

A vapor possessing a water content of 0.160 per cent is in equilibrium with a 54.1 per cent ammonia solution at 32 deg. F. Applying the preceding, this same vapor will be in equilibrium with a 50.9 per cent solution at 6 deg. F. The value of the quotient corresponding to a solution of this concentration is 0.903. Knowing the temperature of the solution in the coils and the quotient, the value of Θ may be calculated, and is found to be 421.0 degrees or a Fahrenheit temperature of $-$ 39.0. The tables of the Bureau of Standards show a pressure of 10.72 lb. of ammonia at this temperature, which will be reached by a 35 per cent ammonia solution at a temperature of 51.3 degrees. In other words, since at this temperature the composition of the vapor required in the coils is identical with the composition of the solution furnished at the expansion valve, there should be no accumulation of a liquid phase in the freezing coils; and, of course, the same condition is true for any absorber temperature below this, provided only that the process occurs at constant pressure.

On the other hand the tables of the Bureau of Standards give the value of 35.09 lb. for anhydrous ammonia at 6 deg. F. This is the pressure attained by a 35 per cent ammonia solution at 105.9 deg. F. and at this pressure the solution phase is identical in composition with that furnished by the expansion valve. No evaporation of water will therefore occur in the freezing coils. In order that the machine may work properly the average temperature of the absorber must be kept below the value of 105.9 deg. F.

We may assume that the absorber temperature is that of the solution leaving the absorber, namely, 80 deg. F. The pressure is 20.5 lb. and the quotient is now 0.954, corresponding to a concentration of solution of 66.1 per cent at 6 deg. F. The corresponding concentration at 32 deg. F. is 69.3 per cent, according to the suggested method, in equilibrium with a vapor of 0.065 per cent water content. The amount of water appearing as an accumulating liquid phase is evidently represented by the difference between the water content of the liquid phase as supplied by the valve and that of the vapor, or 0.095 per cent. Of the 2.39 lb. of water passing

the valve 1.42 lb. will accumulate in the coils to form 4.19 lb. of 66.1 per cent solution, and the rest will evaporate off into the vapor phase together with the residual ammonia.

Performance.—On the basis of 60,700 lb. of ammonia representing the quantity which must be evaporated to supply a refrigerating effect of 100 tons per day, Table 17*b*, the table of preformance, was constructed.

TABLE 17*a*

TABLE OF PERFORMANCE FOR ABSORPTION REFRIGERATING MACHINE

	Pounds per 2000 Lb. of Vapor	Pounds per 100 Tons of Refrigeration per 24 Hours
Weight of original vapor	2000.00	81,300.0
Containing water	148.00	6,020.0
And ammonia	1852.00	75,280.0
Returned to generator, water	144.8	5,890.0
Returned to generator, ammonia	359.7	14,630.0
Sent to receiver, water	2.39	97.2
Sent to receiver, ammonia	1493.1	60,700.0
Accumulated in coils, water	1.42	57.7
Accumulated in coils, ammonia	2.77	112.6
Condensed by absorber, water	0.97	39.5
Condensed by absorber, ammonia	1490.3	60 582.0

Effect of Water Content.—At this point it may be well to clear up an apparent misconception with regard to the nature of the refrigerating medium in the absorption process. In the compression process anhydrous ammonia itself is the substance affecting the essential heat transfers. In the absorption process this is not the case, but a concentrated solution of ammonia, or better still, a solution of water in ammonia is used as the active agent. This itself in no way affects the value of the machine for producing cold; in fact, as far as can be predicted with our present knowledge of the ammonia solution, this should increase its efficiency since the latent heat of water boiling under any pressure is much higher than that of ammonia at the same pressure. The specific heats of both liquid and vapor of steam and of ammonia are nearly identical. On the other hand, the use of a solution in the coils instead of a pure substance introduces the undesirable effect of selective evaporation which under ordinary working conditions causes an accumulation of a certain amount of liquid phase in the freezing pipes. Probably most of this is eventually carried along to the absorber by the impetus of the vapor current, thus preventing clogging. It must be

remembered that the calculation here appended concerns only constant pressure processes. A pressure drop through the freezing coils will affect these calculations unfavorably by further increasing the amount which will form of this liquid phase.

Furthermore, it is evident now that this condensation does not form water but a concentrated ammonia solution in the coils. Herein lies the explanation of the fact that these machines can have condensation occurring in the coils and yet not suffer plugging. Although no work has been done to determine the freezing points of ammonia solutions, the law regarding the lowering of the freezing point is universally true; and under the conditions of concentration existing in the cooling coil it would be impossible to plug the line with frozen solution unless the back pressure from the absorber were materially reduced.

Operating Conditions.—In the operating conditions which we have just been considering there is another interesting possibility to be investigated. In the analysis it was stated that the vapors passing through the rectifier were cooled to a temperature of 98.81 degrees, or 20 degrees above the condensation point of anhydrous ammonia. It is at once evident that the amount of accumulated liquid in the brine coils will be materially decreased if this rectifying is carried still further; and it may not be out of place to study the operation of a machine in which the vapors leave the rectifier at a temperature of 88.81 degrees or 10 degrees below the first rectifying temperature.

To save space the author has appended Table 17*b* as a means of contrasting the results due to the two operating conditions. The table is self-explanatory. It is evident that in order to secure identical refrigerating effects, the quantity of vapor furnished by the generator must be much greater in the second case than in the first, and a much larger amount of ammonia must be returned to the generator from the rectifier. On the other hand, it is possible to raise the absorber to a higher temperture before accumulation begins in the freezing coils (optimum temperature), and as would be expected the amount of this accumulation is not so great for any given temperature, thus increasing the refrigerating efficiency of the coils. Yet the maximum temperature to which the absorber can be raised without prohibiting the proper functioning of the machine is not altered.

Here again the engineer is confronted by conflicting conditions between which it is better to effect a compromise, but the nature of this compromise must be determined by the results to be obtained. A concentrated ammonia solution boils at a lower temperature than one more dilute; hence it follows that in plants desiring the attainment of a cold temperature fairly close to the bottom of the range capable of being reached with the absorption machine it would be better to keep the rectifier at a lower temperature, and thus furnish the freezing coils with the more concentrated ammonia. In those localities in which the climate is such that the cooling water around the absorber does not keep its temperature very low, it would greatly increase the working efficiency of the machine to supply only the more concentrated ammonia to the coils; but otherwise there seems to be no practical advantage in keeping the rectifier at this lower temperature, in view of the heavier load which is thus transferred to the generator.

The chart, Fig. 60, will be found very useful in the solution of engineering problems. From it values required for computations can be found directly, accurately enough for most work. In addition a clearer idea of conditions can generally be obtained from a chart than by the use of tables, and this is particularly true in the case of the absorption machine cycle.

TABLE 17b

COMPARATIVE PERFORMANCE FOR DIFFERENT CONDITIONS OF OPERATION

Items	Per Ton of Vapor	Per 100 Tons of Refrigeration per 24 Hours	Per Ton of Vapor	Per 100 Tons of Refrigeration per 24 Hours
Weight of original vapor (pounds)	2000.0	81,300	2000.0	115,200
Weight of contained water	148.0	6,020	148.0	8,500
Weight of contained ammonia	1852.0	75,280	1852.0	106,700
Temperature of vapor leaving rectifier (degrees F.)	98.81		88.81	
Concentration of solution returned to generator (per cent)	71.30		84.50	
Weight returned (pounds)	504.5	20,530	946.0	54,490
Weight of water	144.8	5,890	146.6	8,440
Weight of ammonia	359.7	14,630	799.4	46,050
Water content of vapors from receivers (per cent)	0.160		0.065	
Weight of this vapor (pounds)	1495.5	60,800	1054.0	60,740
Weight of water	2.39	97.2	0.69	40
Weight of ammonia	1493.1	60,700	1053.3	60,700
Maximum absorber temp. (degrees F.)	105.60		105.60	
Maximum absorber pressure (pounds)	35.09		35.09	
Optimum absorber temp. (degrees F.)	51.30		79.50	
Optimum absorber pressure (pounds)	10.72		20.34	
Pressure of absorber at 80 deg. F. (pounds per square inch)	20.50		20.50	
Composition of equilibrium solution at 6 deg. F. (per cent)	66.10		66.10	
Water content of vapor at 6 deg. F. (per cent)	0.065		0.065	
Water content accumulating in coils per cent)	0.095		0.00	
Weight of accumulating solution (pounds)	4.19	170.3	0.0	0.0
Weight of water	1.42	57.7	0.0	0.0
Weight of ammonia	2.77	112.6	0.0	0.0
Weight of vapor condensed in absorber (pounds)	14.9130	60,621.3	1054.0	60,740.0
Weight of water	0.97	39.5	0.69	40.0
Weight of ammonia	1490.30	60,581.8	1053.3	60,700.0

TABLE 18

Total Vapor Pressures of Aqua Ammonia

Pressures are in Pounds per Square Inch Absolute

Temp. Deg. F.	Molal Concentration of Ammonia in the Solutions in Percentages										
	0	5	10	15	20	25	30	35	40	45	50
32	0.09	0.34	0.60	0.97	1.58	2.60	4.20	6.54	9.93	14.18	19.40
40	0.12	0.45	0.77	1.24	2.01	3.25	5.21	8.06	12.05	17.20	23.39
50	0.18	0.64	1.05	1.65	2.67	4.29	6.75	10.35	15.34	21.65	29.26
60	0.26	0.86	1.42	2.21	3.51	5.55	8.65	13.22	19.30	27.05	36.26
70	0.36	1.17	1.84	2.90	4.56	7.13	11.01	16.56	24.05	33.39	44.42
80	0.51	1.52	2.43	3.76	5.85	9.06	13.86	20.61	29.69	40.96	54.08
90	0.70	2.02	3.15	4.83	7.43	11.40	17.23	25.48	36.34	49.82	65.32
100	0.95	2.62	4.05	6.13	9.34	14.22	21.32	31.16	44.12	59.99	78.30
110	1.27	3.34	5.14	7.72	11.64	17.58	26.07	37.81	53.16	71.87	93.19
120	1.69	4.27	6.46	9.63	14.42	21.54	31.69	45.62	63.59	85.33	110.20
130	2.22	5.38	8.07	11.91	17.67	26.20	38.25	54.55	75.55	100.86	129.50
140	2.89	6.70	9.98	14.63	21.49	31.54	45.73	64.78	89.19	118.24	151.30
150	3.72	8.29	12.23	17.81	26.00	37.81	54.43	76.61	104.65	138.10	175.40
160	4.74	10.16	14.92	21.54	31.16	45.02	64.25	89.88	122.10	160.20	202.70
170	5.99	12.41	18.01	25.87	37.11	53.27	75.55	104.84	141.75	185.10	233.20
180	7 51	15.00	21.65	30.86	44.02	62.68	88.17	121.68	163.70	212.60	267.00
190	9.34	18.06	25.87	36.60	51.81	73.32	102.56	140.75	188.10	243.30	304.30
200	11.53	21.60	30.72	43.14	60.62	85.33	118.68	161.81	215.20	277.00	345.50
210	14.12	25.61	36.26	50.58	70.72	98.80	136.42	185.10	245.10	314.50	390.70
220	17.19	30.27	42.47	59.00	81.91	113.81	156.41	211.24	278.20	355.10	439.60
230	20.78	35.59	49.60	68.46	94.43	130.64	178.28	239.70	314.50	400.20	493.40
240	24.97	41.52	57.65	78.91	108.60	149.20	202.74	270.92	354.10	448.90	552.30
250	29.83	48.32	66.67	90.74	124.08	169.48	229.62	305.60	397.60	502.40	

TABLE 18—*Continued*

TOTAL VAPOR PRESSURES OF AQUA AMMONIA

Pressures are in Pounds per Square Inch Absolute

Temp. Deg. F.	Molal Concentration of Ammonia in the Solutions in Percentages									
	55	60	65	70	75	80	85	90	95	100
32	25.16	31.16	36.77	42.72	45.94	49.28	52.14	54.90	58.01	62.29
40	30.20	37.20	43.73	49.60	54.43	58.33	61.64	64.78	68.32	73.32
50	37.54	45.93	53.85	60.87	66.67	71.29	75.25	79.07	83.41	89.19
60	46.23	56.32	65.90	74.06	80.96	86.49	91.08	95.69	100.66	107.60
70	56.44	68.46	79.54	89.36	97.51	104.08	109.60	114.86	120.63	128.80
80	68.19	82.55	95.69	107.20	116.54	124.30	130.64	136.40	143.72	153.00
90	81.91	98.61	114.02	127.42	138.34	147.15	154.56	161.81	169.76	180.60
100	97.68	117.17	135.01	150.50	163.16	173.40	182.10	190.22	199.22	211.90
110	115.70	138.10	158.84	176.54	191.15	203.26	212.89	222.34	232.85	247.00
120	136.20	162.08	185.70	206.29	222.68	236.37	247.38	258.40	270.10	286.40
130	159.40	189.00	215.88	239.33	258.40	273.30	286.40	298.67	311.90	330.30
140	185.40	219.28	249.66	276.15	297.81	315.00	329.40	343.20	358.60	379.10
150	214.50	252.65	287.24	317.30	341.70	361.10	377.10	392.80	409.80	433.20
160	247.00	290.18	329.40	363.10	390.20	412.20	430.40	447.80	466.60	492.80
170	283.10	331.70	375.60	413.30	443.70	467.80	488.70	508.20	528.80	558.40
180	323.10	377.10	426.60	468.40	502.40	529.50	552.30			
190	367.10	427.70	482.50	528.80						
200	415.10	483.00	543.60							
210	468.40	542.90								
220	525.50									
230										
240										
250										

Figures 55 to 58 show details of the absorption machine, and Table 16 gives the surfaces used in standard machines. Also Table 18 gives the total vapor pressure of aqua ammonia and Fig. 60 the partial pressures

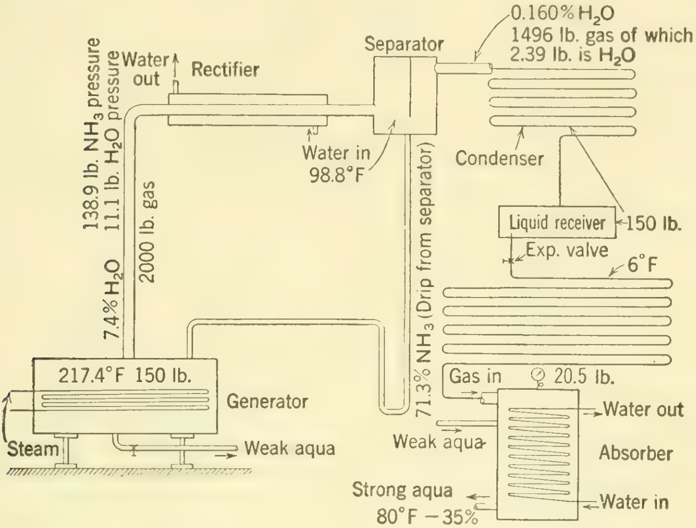


Fig. 61.—Schematic Arrangement of the Absorption Machine.

of water vapor above aqua ammonia. These values and the chart are taken from Bulletin No. 146 of the Engineering Experiment Station of the University of Illinois.

The Absorber.—The action of the absorber is a complex one, not easily defined in a single statement. The gas from the evaporator enters the absorber and comes in contact with a spray of weak, cooled aqua, and a strong aqua is formed. For simplicity the process in the absorber is divided into three parts, namely, *a*, the cooling of the weak aqua to the temperature of the strong aqua leaving the absorber, *b*, the condensation of the gas from the evaporator, and *c*, the absorption of the liquid ammonia by the weak aqua. The pressure in the evaporator is determined by that in the absorber which in turn is a function of the temperature and concentration of the strong aqua in the absorber. The process of liquefaction and of solution is therefore at constant pressure,

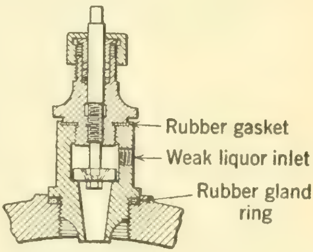


Fig. 62.—Mixing Device in Absorber.

but the temperature of liquefaction (the temperature of anhydrous ammonia corresponding to the pressure) is much less than that of the strong aqua. In consequence the liquid ammonia formed by the liquefaction previous to the entering into solution must be conceived to be *heated* up to this temperature before solution takes place.

In order to bring out the method of calculation the following problem will be solved:

Problem.—The strong aqua leaving the absorber has a concentration of 35 per cent, and a temperature of 80 deg. F. The weak aqua has a concentration of 30 per cent and enters the absorber at a temperature of 120 deg. F. The heat absorbed by the cooling water is required and the number of pounds of water per minute, assuming a temperature rise of 10 deg. F.

The number of pounds of *weak* aqua circulated (Table 15) in order to absorb 1.0 lb. of ammonia gas is given by the formula,

$$n = \frac{1 - x_2}{x_1 - x_2} - 1.0 = \frac{1.0 - 0.30}{0.35 - 0.30} - 1.0 = 13.0 \text{ lb.}$$

where x_1 is the concentration of the strong, and x_2 that of the weak aqua.

The pressure in the absorber, at 35 per cent concentrations and 80 deg. F. is 20.5 lb. The weak aqua is 30 per cent ammonia and 70 per cent water. It has been the practice to assume that the specific heat of the aqua is found by assuming that simply a mixture existed. The specific heat of the liquid ammonia in this case is 1.175, therefore the specific heat of the mixture is

$$(0.30 \times 1.17) + (0.70 \times 1.0) = 1.053$$

a. The cooling effect necessary to cool the weak aqua per 1.0 of ammonia.

$$Q = Mc(t_2 - t_1) = 13.0 \times 1.053 \times (120 - 80) = 547.6 \text{ B.t.u.}$$

b. Cooling required for the liquefaction, etc.

$$Q = 606.5 - 132.0 = 474.5 \text{ B.t.u.}$$

c. Cooling required to overcome the heat of solution. This is expressed by

$$\begin{aligned} Q &= 345(1 - x) - 400x^2 \text{ where } x \text{ is the average concentration} \\ &= 345 \times 0.675 - 400 \times (0.325)^2 = 190.8 \text{ B.t.u.} \\ &\text{Total} \qquad\qquad = 1212.9 \text{ B.t.u.} \end{aligned}$$

The amount of water required with a 10-degree rise of temperature will be

$$1212.9 \div 10 = 121.3 \text{ lb.} = 14.56 \text{ gallons}$$

If a standard condenser pressure was carried in the condenser the ammonia required per ton of refrigeration would be—at 20.5 lb. and 86 deg. F. condensing temperature

$$200 \div (606.5 - 138.9) = 0.428 \text{ lb.}$$

and the water per ton per minute required in the absorber would be

$$14.56 \times 0.428 = 6.23 \text{ gallons}$$

The Munters-Platen Absorption System.—The Munters-Platen system, shown in Fig. 63, is an attempt to eliminate the strong aqua pump. The pressure is exerted throughout the entire system and no "expansion" valve is required. Circulation is obtained by the action of the heating element in the generator. The outstanding feature of the system is the use of an inert gas (hydrogen) in the evaporating coil; sufficient so that the *partial pressure* of the ammonia will be that required by the conditions of operation.

While this system is ingenious it requires several constant conditions for success. The total pressure must remain constant[6] and there must be no *leakage* of hydrogen. As most refrigerating machines have a variable condenser pressure depending on the amount and the temperature of the condensing water it is not clear how successful the Munters-Platen system will be. It undoubtedly will be limited, under any circumstances, to the gas-fired machine.[7]

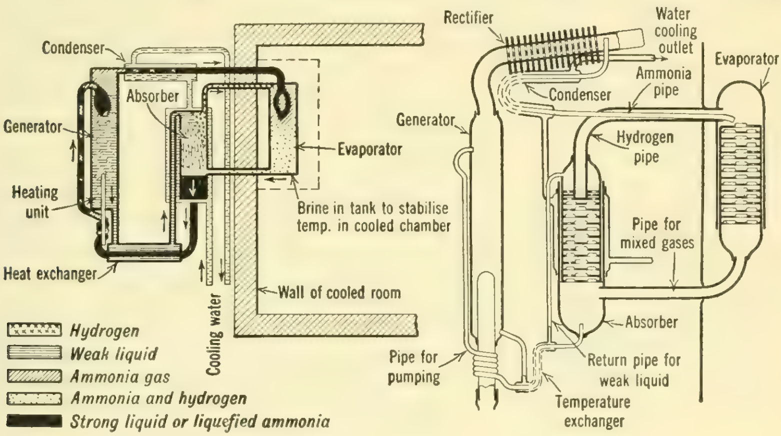


FIG. 63.—The Munters-Platen System.

[6] The pressure in the system may be taken as p lb. per square inch, which is equal to $p_1 + p_2$ in the evaporator coils where:

p_1 = the pressure exerted by the ammonia
p_2 = the pressure exerted by the hydrogen

Suppose the condenser pressure p increases, due to a temperature rise of the condensing water. Then p_1 will increase a like amount as the change of pressure of the hydrogen would be small. The temperature of evaporation would be regulated then by the condenser pressure, a condition which can be controlled to some extent, but usually is not, except to make the condenser pressure as low as possible.

[7] A modification of the Munters-Platen system is the Altenkirch refrigerating machine. This device uses sulphuric acid as a solvent (La Technique Moderne, July 1, 1926).

The Intermittent Absorption Machine.—The intermittent absorption machine is of particular interest for the small sizes, as the chief feature of the design is the elimination of the strong aqua pump and therefore the necessity for the use of power in the machine. It is automatic as regards the water control and the heating element, which is gas fired always because of the cost, and a part of the machine acts both as a generator and an absorber. Quite naturally refrigeration cannot be continuous. However, several large installations have been made, as for example in milk cooling after pasteurization.

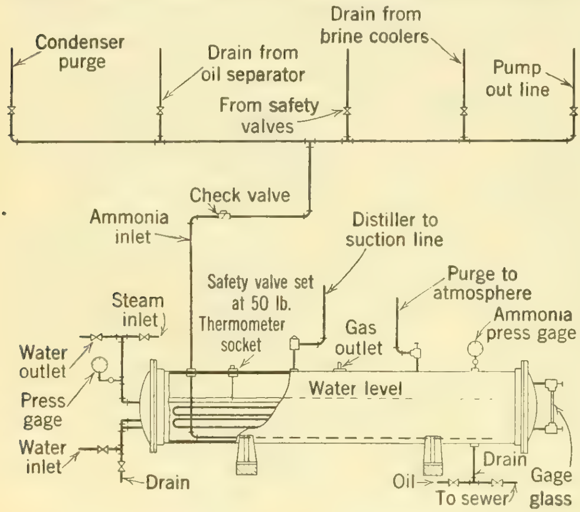


FIG. 64.—Purge Drum for Absorption Machine.

One of the essential features of the absorber is the thorough mixing of the gas and weak aqua. This is accomplished in Fig. 62 by the device of the weak aqua nozzle which forms a spray and permits the aqua in this condition to meet the ammonia gas from the evaporating coils. Another mixer is the double pipe with the inner pipe perforated. The gas passing through the holes in the inner pipe as a series of bubbles meets the liquid in the outer pipe.

Purging.—Figure 61 is a special device adapted to large plants to remove the non-condensible gases. Some disintegration occurs in the absorption machine and air enters in one manner or another, so that purging of the absorber is necessary. Purging always means a loss of ammonia unless a special device is used which in this case is a special absorber generator. As an absorber, water cooled, the ammonia in the purge gas is absorbed and the inert gas is blown off. As a generator the steam coils drive off the ammonia in the still where it is discharged into the condenser.

CHAPTER IV

FITTINGS AND CONDENSERS

Ammonia Fittings.—Because of the nature of ammonia it is necessary to make especial provision against leaks, even though the pressures are not more than 200 lb. per square inch as a rule. The accepted form of flanged joint is that kind employing the tongue and groove, using generally a rubber or lead gasket, although asbestos and other material are sometimes used. Fittings may be designed with screw ends—up to 3 to 5 in.,—with oval, square and round flanges, depending on the size of the pipe.

There are two methods of making up pipe and fittings,[1] the first method being the litharge and glycerine, and the second being the solder joint in the case of black iron and steel pipe. As a rule the latter is confined to condensers and (possibly) to brine coolers that are made up at the factory, and where the necessary equipment is at hand for the purpose. The process is to tin the pipe—using a bright pipe and screw threads and an acid composed of muriatic acid that has been partly neutralized by the action of zinc—and then to make up the joint with the pipe and the fitting heated up to the temperature of the solder. In the larger sizes of fittings there are solder recesses (Fig. 65) to be filled with solder and this is caulked in order to cover the threads. According to Fairbanks[1a] the defects of soldering are two in number. First, in the larger pipes—3 in. and over—vibration, expansion and contraction of the pipe finally break the soldered joint with the result that a leak develops. Second, the temperature of the discharge connections from the compressor is often 300 deg. F. in which

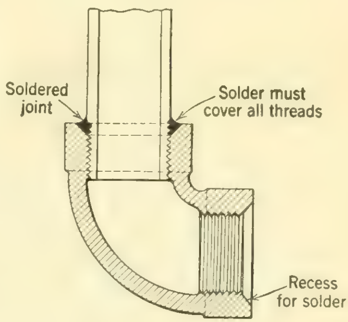


Fig. 65.—Fitting with Solder Recess.

[1] For a more complete discussion of erection problems see Chapter X.

[1a] Fairbanks in the Mechanical Engineers Handbook, McGraw-Hill Book Company, p. 1729.

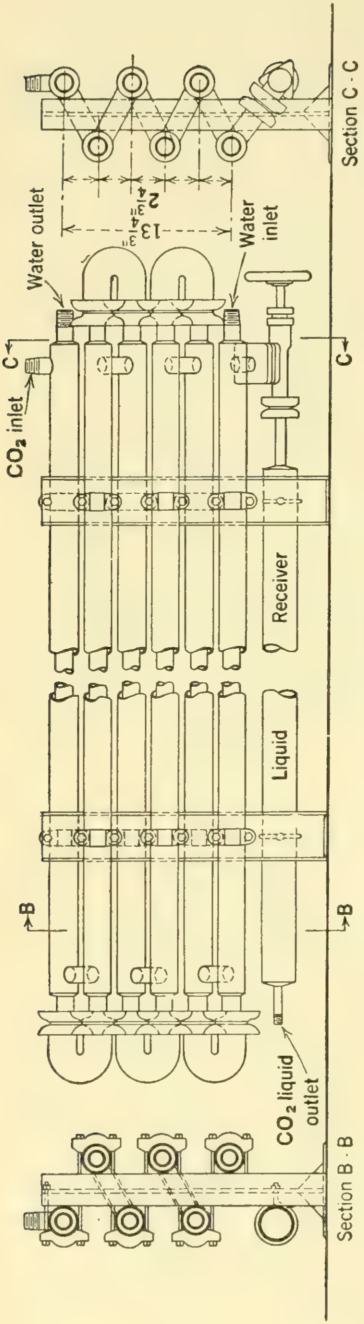


Fig. 66.—Double Pipe Welded CO_2 Condenser.

case the solder becomes plastic. Fairbanks favors flanging and V grooving the flange for all pipe 3 in. and over, in which case full-weight pipe only is used, employing the Van Stone joint and an antimonial lead gasket.

The essential feature of the ammonia pipe connection is a clean and accurate thread. The amount of pull or the heat developed during the making up of the threaded joint is not a reliable evidence of a good pipe connection as these factors could be caused by dirt. In order to secure good work the threads must be cleaned carefully, preferably by means of gasoline, and assurance obtained that chips, and other loose pieces of steel, are not caught in the threads. It is usual to pull up on the pipe threads much more than for similar work in steam, and the result is that all screw fittings are much heavier than those designed for water or steam for 250 lb., and in addition the cast iron of the fittings must be of a dense structure and always heavier-walled than a steam fitting would be. For some time the close-grained, air furnace, gray iron casting was used but this is being replaced by the semi-steel and the drop-forged steel fitting on the part of some of the manufacturers. These fittings preferably should be malleable for screwed and semi-steel for flanged fittings and drop forged for flanges. The drop-forged steel fitting is becoming more popular, although more expensive, be-

cause of its lack of porosity, its great strength and the fact that it can be welded to the pipe. This last property is not so very important with ammonia machines but very desirable in connection with carbon dioxide refrigeration.

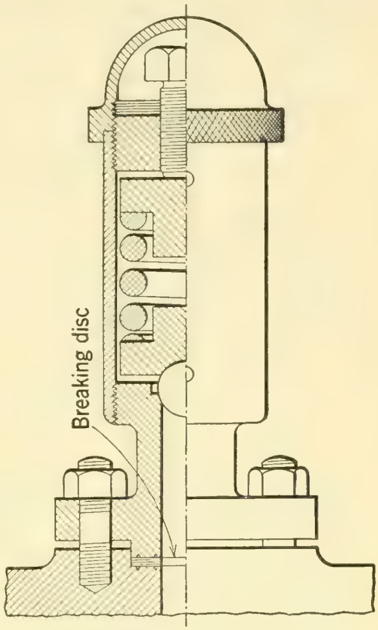


Fig. 67.—Safety Valve for Carbonic Compressors

In ammonia work it has been found best to use a soft gasket, and to form the joint by a method which will prevent the gasket from squeezing out. The tongue and groove design absolutely prevents the blowing out of the gasket material, and the soft lead asbestos gasket seems to give good service under these conditions. The ordinary copper and asbestos gasket—with the flat-faced flange—is never attempted, although the Fairbanks design is somewhat like the flat-faced flange, both on account of inability to secure a sufficient unit pressure on the packing and also because the asbestos will flatten and the copper will corrode (if water is present in the ammonia) and flatten when under ammonia pressure. The present tendency is to eliminate valves and fittings as far as possible by using flame, electric and forge welding at the shop and flame and thermit welding on the job. Pipe work can be made welded entirely except for the unions and perhaps the stop and the expansion valves. Some double pipe condensers are made—especially for carbon dioxide or for small capacity plants—welded throughout (Fig. 66); and apparently welding will supersede fittings whenever the cost as well as the conservation of the refrigerant makes this justifiable. For evaporator pipe coils the flanged bent pipe is rapidly replacing the cast return bend (Fig. 67) except when the pipes are submerged in a liquid, in which case continuous welded pipe is used. It cannot be said, however, that there is any great tendency to do away with the double pipe and the atmospheric type of condenser for ammonia, both of which are plentifully supplied with fittings and valves.

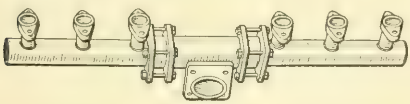
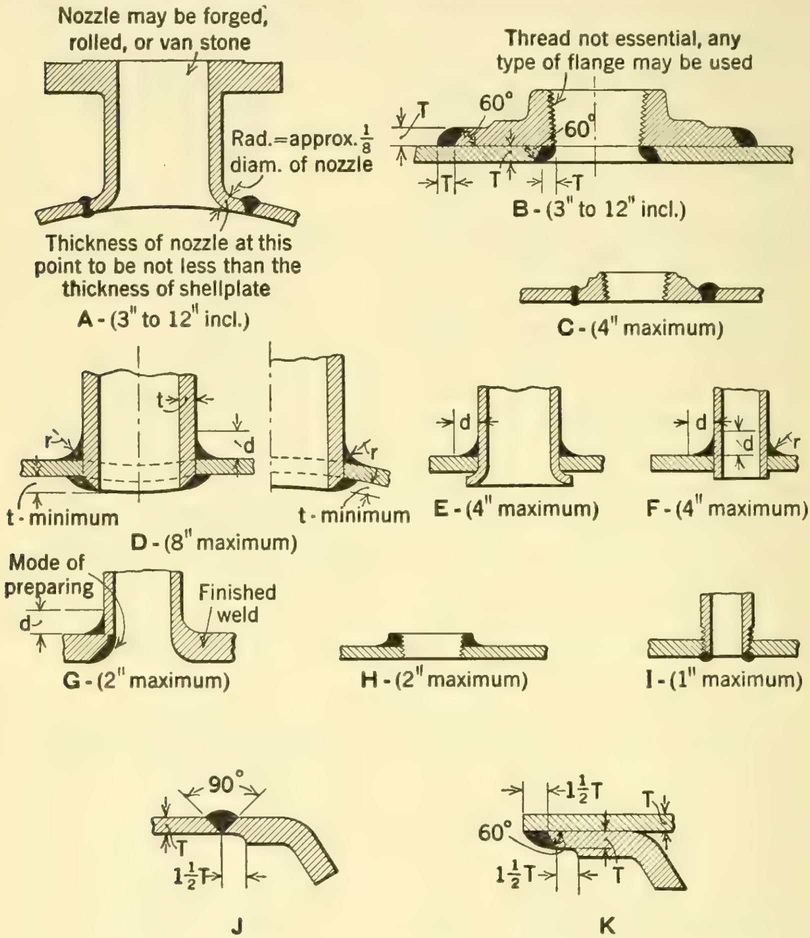


Fig. 68.—Welded Headers.

Manifolds and Headers.—Manifolds and headers are made either of semi-steel castings or are made up of steel and welded, for which purpose the autogenous (flame) weld is very successful (Fig. 68). This



Note:- For all nozzles, dimensions d and r shall not be less than the following:

$$d = \frac{\text{Nominal pipe size}}{8} + \frac{1''}{4} \qquad r = \frac{\text{Nominal pipe size}}{2}$$

FIG. 69.—Methods of Flame Welding.

is the method whereby the gases in two tanks, one of acetylene (C_2H_2) and the other of oxygen, are mixed in a nozzle and the mixture is controlled by separate valves. The weld is made by fusing the material

to be welded and by melting a steel rod until the proper reinforcement is obtained (Fig. 69).[2] The flame weld has not been entirely successful, and for certain work the forge weld has taken its place, as for example in the welded heads of liquid receivers. Also for the same reason the flame weld along the longitudinal seam is being done away with, wherever convenient, by the use of the proper sized lap welded pipe manufactured in the steel mill, and pipe up to 90 in. diameter is obtainable.

Valves.—Valves for ammonia are usually of the globe and the angle type. When used as stop valves they may be furnished with either

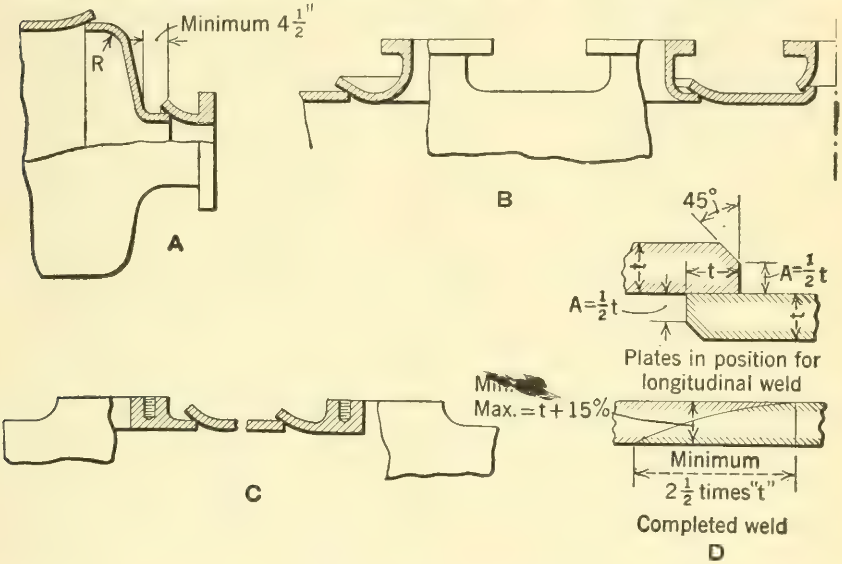


FIG. 70.—Methods of Flame Welding.

hard or soft seats. The advantage of the soft, white metal seat—dovetailed into the valve disc—is that foreign material, like steel cuttings, will sink into the soft material instead of preventing the valve from seating and therefore causing a leak, but the soft seat has to be renewed at regular intervals. Usually the valve is so designed as to make a seat at the *top* of the valve when it is full opened (Fig. 71) in order that leakage past the stem through the packing when the valve is open may be prevented and also so that the valve stem may be repacked while the valve is under pressure.

Special designs of regulating (expansion) valves are numerous. These have a special fine thread on the spindle for fine adjustments,

[2] Rules for the construction of unfired pressure vessels, A. S. M. E. Boiler Construction Code.

and are made with either a conical projection of the spindle through the valve seat, or a cylindrical projection with a tapered groove in order to obtain the required close adjustment. When closed the expansion valve (Fig. 71) can be made tight by means of a soft metal seat ring or by means of a plain needle-type valve of hard steel. When such close adjustments are not required a modified globe or angle valve may be used.

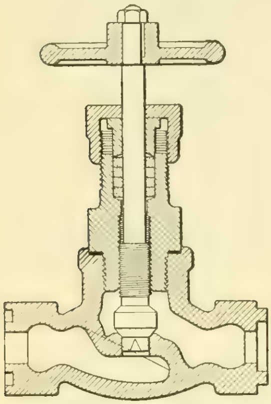


Fig. 71.—Section of an Expansion Valve.

The expansion cock is still used but it is inclined to jam, and at the present time it is not very popular. The flow of one cubic foot of liquid ammonia per minute is sufficient for approximately 100 tons of refrigeration, and with a difference of pressure of 150 lb. per square inch on the two sides of the valve a large opening is not required. On the other hand with the use of the expansion cock the suction pressure is harder to regulate than with the usual expansion valve.

Figures 72 to 85 show various forms of fittings. Figure 80 is one design of strainer and Fig. 81 one form of an ammonia purifier. Figures 83, 84 and 85 and Table 19 give dimensions of some of the standard large accessories. An isometric sketch of the low-pressure suction return to the compressor, showing pipes, fittings and an accumulator is given in Fig. 86. Tables 22 to 31 give dimensions of the more common

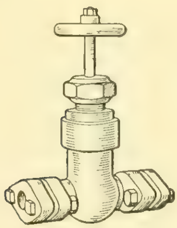


Fig. 72.—Expansion Valve.

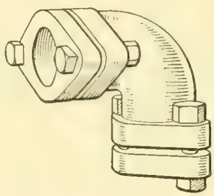


Fig. 73.—Elbow.

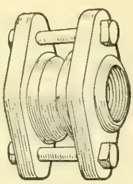
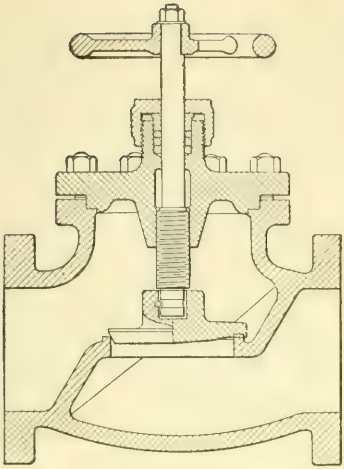


Fig. 74.—Flange.

York fitting, Tables 32 to 35, inclusive, give Frick practice, Tables 36 and 37 show Arctic, whereas Tables 20 and 21 give the dimensions of Crane flanges. It will be noticed on comparing these tables that even the flanges for ammonia are not standard either as regards the diameter, the width or the depth of the groove, and even the thickness of the metal, size and spacing of the bolts, the distance between faces, etc.



bear no resemblance to one another in the different makes. Very unfortunately there does not appear at present any possibility of standardization. The Crane fitting dimensions for ammonia are taken from what is known as "extra heavy steam," and the metal thicknesses,

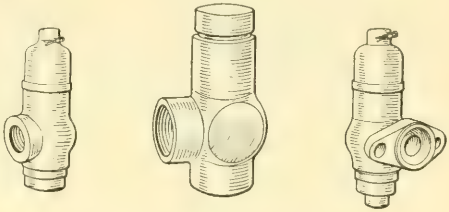


FIG. 75.—Section of Globe Valve.

FIG. 76.—Pop Safety Valves.

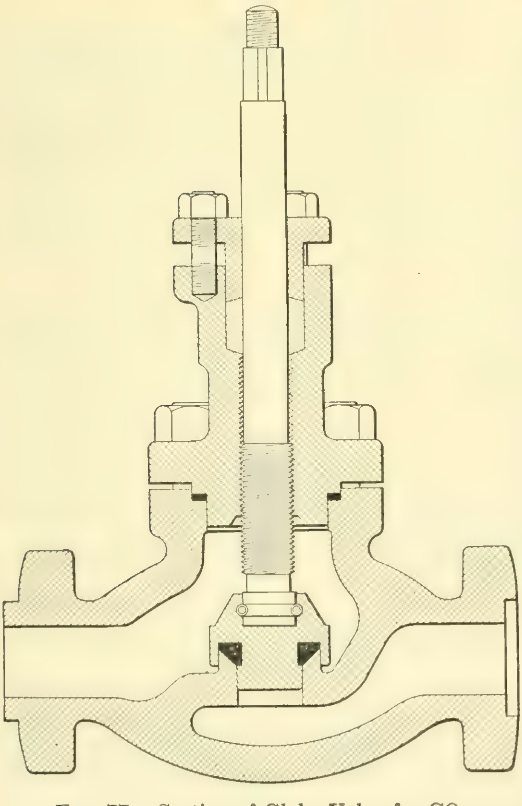


FIG. 77.—Section of Globe Valve for CO_2.

the size of flanges and boltings and the size of the bolts are the same as for their steam fittings and valves.

Carbon Dioxide Fittings.—Figures 87 and 88 show typical carbonic fittings. Figure 88 shows a flange joint for a large pipe wherein the gas seal is made by means of lead gaskets made tight by means of an annular gland. In the smaller sized pipe flanges the joint has been made by permitting the pipe (the ends of which have been reamed square) to project $\frac{1}{32}$ in. beyond the flange, and a hard red fiber gasket has been used to make up the joint. However the hard fiber has a

tendency to crack and later practice has been in the direction of a composition aluminum or copper gasket. Figure 88 shows a typical $\frac{1}{2}$ by $1\frac{1}{4}$ expansion valve which is noteworthy only on account of the heavy metal construction.

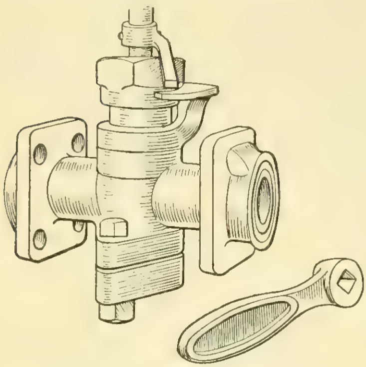


FIG. 78.—Expansion Cocks.

Carbonic fittings are not standard and each manufactory has its own patterns. It is becoming more and more common to make use of drop-forged fittings for carbon dioxide or, more generally speaking, for the line of fittings designed for 3000 lb.

CONDENSERS

Historical.—In the early days of refrigeration the condenser was nothing else than a round or a flat coil of pipe made up with suitable fittings at first and then, as time elapsed, the pipe was welded completely or was made up with a combination of welding and fittings. The refrigerating engineer was obsessed with the idea of the danger due to ammonia and the difficulty in reducing ammonia leaks, the latter being important in the last quarter of the nineteenth century because of the heavy cost. The refrigerating engineer felt that he could not take chances of rupture, but he knew that the pipe condenser could be made safe and reasonably tight. The submerged and the atmospheric condenser, made up with pipe and fittings, were the result, and later the shell and coil condenser was developed.

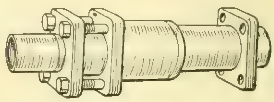


FIG. 79.—Expansion Joint for Ammonia.

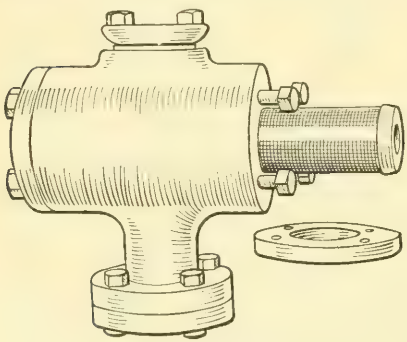


FIG. 80.—Strainer.

As time elapsed, and with the resulting experience and improved materials available for use, the refrigerating engineer devised more complicated fittings which resulted in the double-pipe counterflow condenser and later still this was modified for carbon dioxide refrigera-

tion by welding throughout. Finally came the shell and tube condenser, similar in construction to the steam surface condenser, but with the tubes rolled into both of the tube sheets. At first the shell was

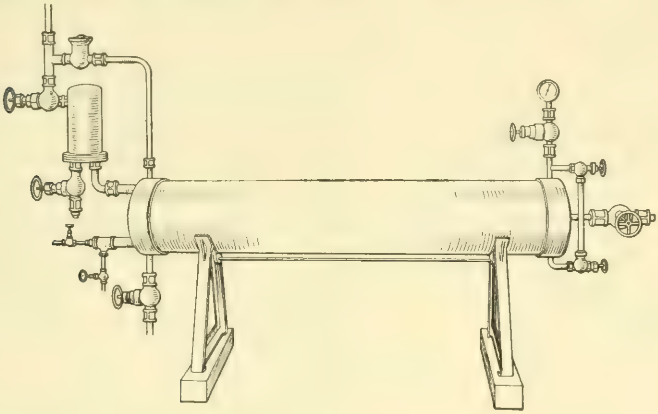


Fig. 81.—Ammonia Distiller.

riveted as well as welded, but the practice in 1926 was usually to take the lap-welded pipe as manufactured at the steel mills. This condenser was designed for vertical and horizontal shells, but the vertical is preferred and very nearly the only one used in the larger sizes.

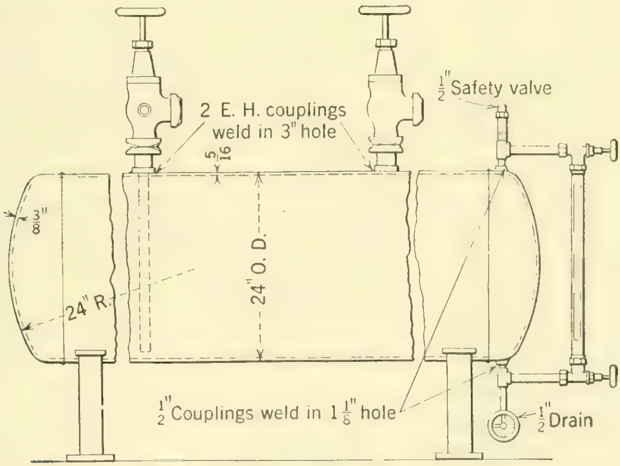


Fig. 82.—Ammonia Liquid Receiver.

As compared with steam engineering it would seem that the progress in the development of the ammonia condenser has been very slow. The

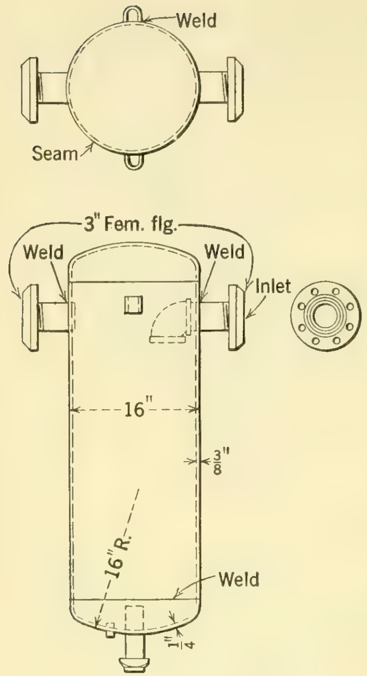


FIG. 83.—Accumulator.

pipe condenser appears to be too complicated, and to be unnecessarily supplied with valves and fittings. In addition the space occupied is excessive at times, and certain of the designs are difficult to arrange so that the different stands will have an equal amount of the load. It seems at first as if the design of condenser made up of pipes and fittings must be inefficient when one considers that it is, in all intents and purposes, a long pipe which may have a dead end or is connected to a liquid receiver, a design which has hardly a counterpart in steam engineering.

On the other hand the pipe designs of condensers have given good service, are reasonably low in first cost and have been economical in the use of water. They can be erected so that the leakage loss is very small and repairs (if such are required) can be made without great difficulty.

Function of the Condenser.—The condenser is for the purpose of removing the superheat and the latent heat of liquefaction, and except in very special cases, it is water cooled. The temperature of the gas entering the condenser may be as high as 250 deg. F. or even 300 deg. F. and the amount of heat removed varies with the operating conditions from 220 to 260 B.t.u.

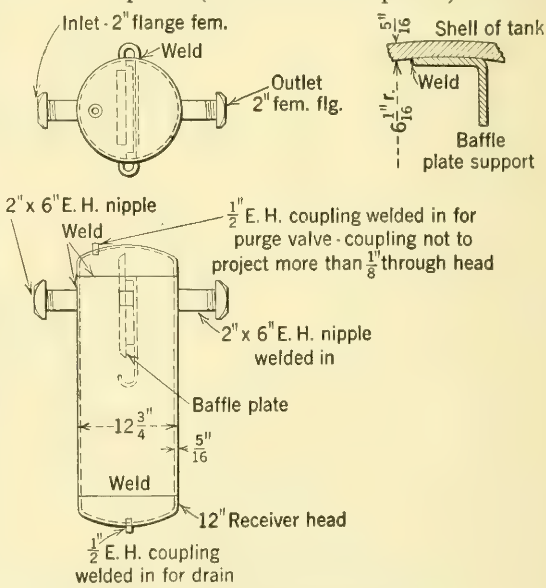


FIG. 84.—Oil Separator.

per ton per minute. Some designs function partly as liquid receivers, particularly in the case of carbon dioxide, but this is a minor object of the condenser and one which is being gradually discarded as a characteristic of the apparatus. The main object is

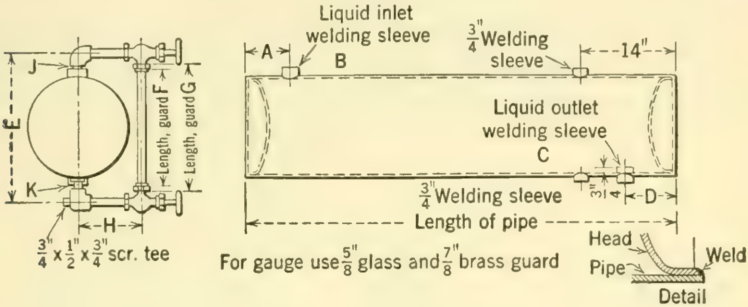


FIG. 85.—Liquid Receiver.

Diameter of Pipe	Length of Pipe		Contents, Cubic Feet	Thickness		A	B	C	D	E	F	G	H	J	K
				Of pipe	Of heads										
	Ft.	In.													
10	8	0	4.4												
10	10	0	5.5	$\frac{3}{8}$	$\frac{1}{2}$	8	$1\frac{1}{4}$	1	8	17	$11\frac{3}{4}$	14	$6\frac{1}{2}$	$1\frac{3}{8}$	2
10	14	0	7.7												
10	18	0	9.9												
12	15	0	11.8												
12	17	6	13.8	$\frac{3}{8}$	$\frac{1}{2}$	8	$1\frac{1}{2}$	$1\frac{1}{4}$	8	19	$13\frac{3}{4}$	16	$7\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{3}{8}$
12	20	0	15.7												
16	15	0	19.0												
16	17	6	22.2	$\frac{3}{8}$	$\frac{5}{8}$	8	2	2	8	23	$17\frac{3}{4}$	20	$9\frac{1}{4}$	$1\frac{3}{8}$	2
16	20	0	25.4												
20	14	0	27.9												
20	17	0	33.9	$\frac{1}{2}$	1	8	2	2	8	27	$21\frac{3}{4}$	24	$11\frac{1}{4}$	2	2
20	20	0	40.0												
24	16	0	46.2	$\frac{9}{16}$	$1\frac{1}{8}$	8	2	2	8	31	$25\frac{3}{4}$	28	$13\frac{1}{4}$		
24	20	0	57.7												

All dimensions given in inches unless otherwise specified.

to liquefy the gas at as low a temperature and pressure as is possible with the quantity and temperature of cooling water available at the plant. How this pressure varies with different conditions of cooling water is shown in Table 75.

Types of Condensers: *The Submerged Condenser.*—The submerged condenser is an obsolete design for ammonia at the present time except

for marine work and carbonic refrigeration. It consists of a coil or a set of concentric coils placed within the shell or tank. The refrigerant passes through the coils and the water is within the shell or tank outside the pipe coils. The coils must be made up without joints below the water surface and they are preferably welded. When more than one coil is used they are brought together in a header. With carbon dioxide the submerged condenser is still used to some extent, and the design can be made more efficient in heat transfer by the use of special devices to increase the velocity of the water flow past the surface of the pipe

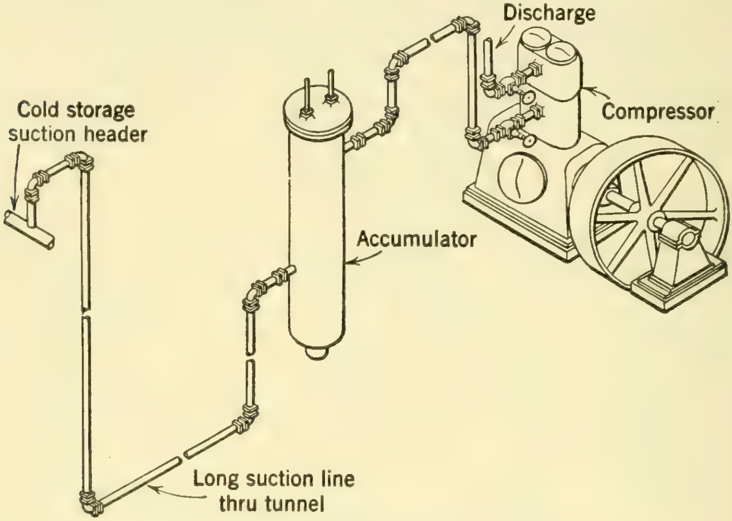
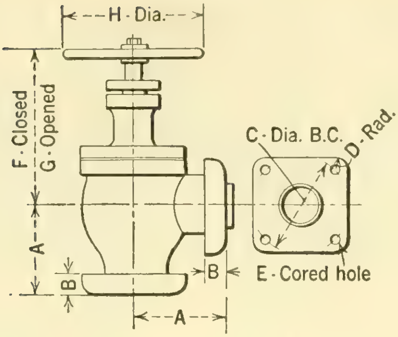


FIG. 86.—Method of Piping an Accumulator.

coils by the use of an inner pipe or by the use of suitable baffles (Fig. 89). In the old designs the rate of heat transfer between the refrigerant and the water is approximately 0.2 of that in the better types of condensers, but the later designs for carbon dioxide have improved the performance considerably, although there is difficulty in cleaning the coils if dirty or scale forming water is used.

The Atmospheric Condenser.—The atmospheric condenser is the general term for that design which is made up in stands with the water delivered to the outside of the upper pipe with the succeeding pipes receiving the water by dripping from one pipe to the next. The gas from the compressor may enter at the top (the old type), at the bottom for about three pipes and then pass to the top (the common atmospheric condenser) or at the bottom (the bleeder type). The British design of atmospheric condenser consists frequently of sheets of pipes, usually

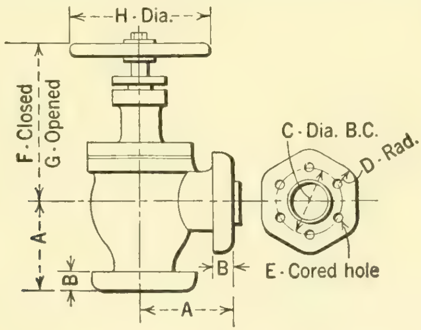


a

Triseal Angle Valve
4-Bolt Type

Size	Symbol Number	A	B	C	D	E	F	G	H	Number of Bolts	Size of Bolts
$\frac{3}{4}$	1206-F	$3\frac{3}{4}$	$\frac{7}{8}$	$3\frac{1}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$9\frac{1}{2}$	$10\frac{1}{4}$	5	8	$\frac{1}{2}\times 2\frac{3}{4}$
1	1202-F	$4\frac{3}{16}$	$1\frac{1}{16}$	$3\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$11\frac{1}{2}$	$12\frac{1}{2}$	7	8	$\frac{5}{8}\times 3\frac{1}{4}$
$1\frac{1}{4}$	1180-F	$4\frac{5}{16}$	$1\frac{1}{8}$	$4\frac{1}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	11	$12\frac{3}{8}$	7	8	$\frac{5}{8}\times 3\frac{1}{4}$
$1\frac{1}{2}$	1240-F	$5\frac{3}{16}$	$1\frac{1}{8}$	$4\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	14	$15\frac{1}{2}$	$9\frac{1}{2}$	8	$\frac{5}{8}\times 3\frac{1}{4}$
2	1232-F	$6\frac{1}{8}$	$1\frac{1}{4}$	$5\frac{1}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$13\frac{1}{2}$	$15\frac{1}{2}$	$10\frac{1}{2}$	8	$\frac{3}{4}\times 3\frac{3}{4}$

All dimensions given in inches unless otherwise specified.



b

Triseal Angle Valve
6-Bolt Type

Size	Symbol Number	A	B	C	D	E	F	G	H	Number of Bolts	Size of Bolts
$2\frac{1}{2}$	1284-F	$6\frac{13}{16}$	$1\frac{5}{8}$	$6\frac{1}{2}$	1	1	$14\frac{1}{2}$	$16\frac{1}{2}$	$10\frac{1}{2}$	12	$\frac{3}{4}\times 4\frac{1}{2}$
3	1286-F	$7\frac{5}{16}$	$1\frac{3}{4}$	$7\frac{3}{4}$	1	$1\frac{1}{8}$	$17\frac{3}{4}$	$20\frac{1}{2}$	13	12	$\frac{7}{8}\times 5$
4	1320-F	$10\frac{3}{4}$	2	$9\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$21\frac{1}{2}$	$24\frac{5}{8}$	13	12	$1 \times 5\frac{1}{2}$

All dimensions given in inches unless otherwise specified.

Fig. 87.—Carbonic Fittings, Angle Valves.

three in number, interlaced so as to be vertically over one another. In this design, called the evaporative open condenser, the number of pipes may be as many as 48 high; and the water, being supposed to be cooled by the air and by evaporation, is recirculated. In America the atmospheric condenser which was at first 30 pipes high has been reduced to 18 and finally to 12 except in certain particular designs and no attempt at recirculation of the water is made unless a separate system of water cooling is provided by means of a cooling tower or by the use of sprays.

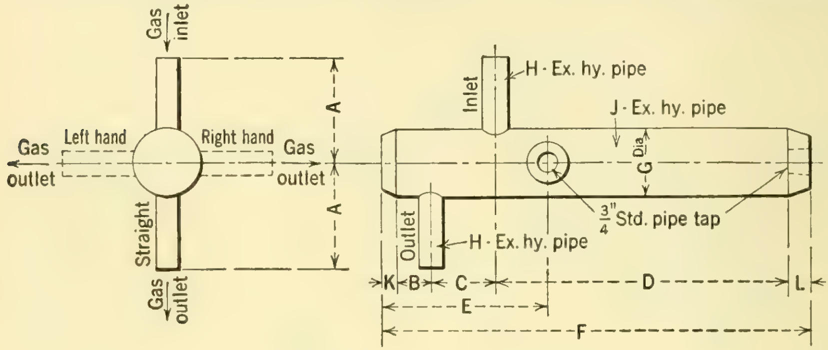


FIG. 88.—Carbonic Fittings, Oil Separator.

WELDED TYPE OIL AND SCALE TRAPS

Size	Symbol Number	A	B	C	D	E	F	G	H	J	K	L
3/4	1321-K	4 1/2	1 1/2	2 1/2	12 1/2	7	18	2 7/8	3/4	2 1/2	5/8	7/8
1	1321-H	4 1/2	1 3/4	2 3/4	16	9	22	2 7/8	1	2 1/2	5/8	7/8
1 1/4	1321-F	5	2 1/4	3 1/4	18 1/2	9	25 1/2	3 1/2	1 1/4	3	5/8	7/8
1 1/2	1321-D	5 1/4	2 1/4	3 1/4	18 1/2	9	25 1/2	3 1/2	1 1/2	3	5/8	7/8

All dimensions given in inches unless otherwise specified.

The design of atmospheric condensers has created considerable interest for a number of years. First came the Block and the Shipley designs of "flooded" condensers whereby the gas from the compressor was mixed with liquid ammonia by means of a sort of ejector nozzle and during the remainder of the process the mixture of gas and liquid was cooled by metal contact with the condensing water as in the other designs of atmospheric condensers. The theory advanced was that the superheat in the ammonia decreased the rate of heat transfer considerably, and that the superheat could be removed quickly by means of this liquid injection. Tests by Ophuls and Greene[3] disproved some of these contentions. Louis Block in the 90's devised a counterflow atmospheric

[3] Ophuls and Greene, American Society of Refrigerating Eng. Journal, 1916.

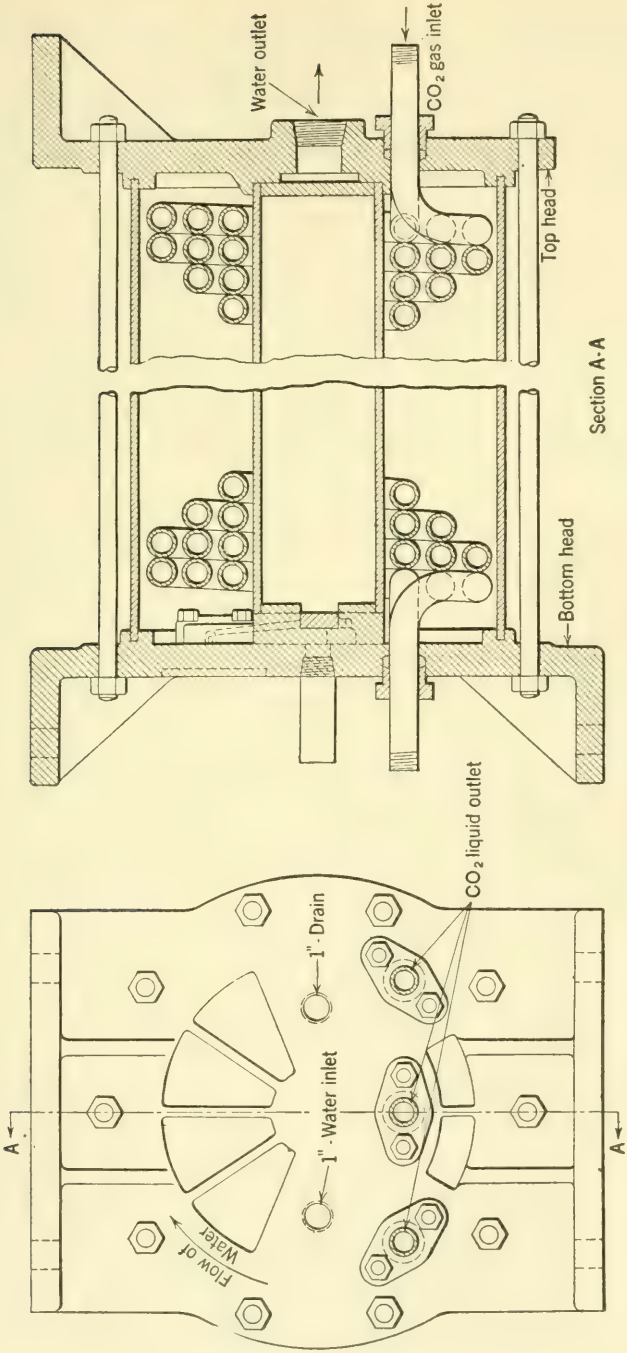


FIG. 89.—Submerged Type CO_2 Condenser.

condenser, bringing in the gas at the bottom and making the upper pipe a dead end. In order to remove the liquid he drained off the condensate at frequent intervals and his condenser became known as the *drip pipe* or *bleeder* condenser. Block's whole design hinged about the importance of securing a counterflow of the refrigerant and the water, an arrangement which had been considered essential for a long time, but which was actually accomplished in the Block (the De la Vergne) design by the use of a large number of fittings. However, recent experiments are changing the opinion as to these condensers, and it is doubtful whether the bleeder or the flooded condenser will be very popular in the future.

Tests in the Engineering Experiment Station of the University of Illinois on a bleeder type of condenser[4] using two thermocouples per pipe for the ammonia temperatures and a carefully designed measuring device for the condensate indicated that from 40 to 70 per cent of all the liquefaction occurred in the upper pair of pipes, the larger amount of condensate occurring at the time that the coldest water was used. During these tests the condenser pressure was constant, and variation was possible by the use of different initial temperatures of the cooling water, and the amount of water used was that required to maintain a constant head pressure. In other words approximately 50 per cent of the gas traveled through some 200 ft. of condenser pipe before liquefaction occurred. Also there was evidence that at certain times the upper pipes filled with condensate so that the load was carried by the next lower pipe. The superheat was found to be removed in the second or third pipe, depending on the suction pressure, which varied the load on the condenser, whereas the fourth, fifth, sixth and the seventh pipe from the bottom carried no appreciable load as they were brought into metal contact with water very near the temperature of liquefaction. The temperature of the ammonia in the condenser was practically constant in the upper nine pipes.

Similar tests on a double pipe condenser[5] indicated very nearly similar conditions to those of the bleeder condenser, except that the maximum liquefaction takes place in the lowest pair of pipes, which are in this case cooled with the coldest water. Here again the temperature of the ammonia was found to be practically constant in the lowest nine pipes. In this case also approximately 50 per cent of the ammonia gas traveled through nearly the entire condenser before liquefaction occurred, and the temperature of the gas during condensation remained constant as theoretical considerations would lead one to expect.

[4] This condenser consisted of 12 2-in pipes, approximately 20 ft. long.

[5] This was $1\frac{1}{4} \times 2$, 12 pipes high, approximately 20 ft. long.

These tests would seem to point out certain fallacies. If the temperature of the ammonia remains constant throughout the condenser where liquefaction occurs there is no material advantage in having a counterflow as the mean temperature between the water and the ammonia will be identical in either case. However, there are designs which utilize the surface more effectively than others. If the superheat is removed in two or three pipes there is no advantage in making the condenser more complex in order to get rid of the superheat. In other words, that part of the condenser exposed to saturated ammonia has drops of liquid forming on the surface, and in consequence the surfaces are wet. In general it has been demonstrated by tests that heat transfer is decidedly affected by the condition of the surface. A layer of inert refrigerant, air or decomposed ammonia only $\frac{1}{100}$ in. thick will decrease the rate of heat transfer to about 1 per cent of the amount that could be obtained if the surface were subject to pure conduction only. The same thing is true of liquids, for if liquids are present a surface film is formed which offers a great resistance to the flow of heat. If the design is such that one half of the cross section of the pipe is filled with liquid that surface not submerged is going to be more effective than the remainder. The bleeder type of condenser, for example, is illustrative of a partially flooded condenser condition as the liquid accumulates until the velocity of gas flow makes the liquid move upwards to a drip connection, or the head is developed sufficient to cause a flow downwards against the flow of the gas. The flooded condenser is designed for more or less filling of the pipes with liquid and the surface not submerged is made highly effective by the increased velocity of the gas flow occasioned by the resulting decrease in the cross-sectional area of the pipe. This velocity is paid for by an increase in the pressure of discharge of the ammonia from the compressor. Tests by Ophuls and Greene and by Shipley[6] have indicated that the *best condenser* is that which drains the condensate most promptly and which has as direct a path from the condenser to the receiver for the liquid as it is practical to obtain. The tests referred to refute absolutely the value of the *flooded type* of condenser of the atmospheric, the double pipe or the shell and tube design and indicate that the value of the bleeder atmospheric condenser cannot be very great from the standpoint of getting counterflow. There may be an advantage, however, in increasing the effectiveness of the heat transfer surface by a prompter removal of the condensate over that which would be possible in the case of the common type condenser and to a lesser extent in the double-pipe counterflow condenser. In all pipe condensers, however, there appears to be an objection due to the

[6] Shipley, National Association of Practical Refrigerating Engineers, 1923.

slowness of the removal of the liquid, and in the resulting non-effectiveness of a portion of the surface.

The Common Atmospheric Condenser.—The common atmospheric condenser is shown in Fig. 90. It usually has at least three pipes at the bottom for the removal of the superheat, and then the gas passes to the top of the condenser. As the top pipe has the coldest water, practically the entire load of condensation is taken by the upper four pipes, except when the condenser has a heavy load due to a higher suction pressure or when relatively large quantities of warm water are showered over the condenser. The liquid produced passes from one pipe to another and finally drains off at the low point. There is small likelihood of liquid sub-cooling because the temperature of the liquid leaving the last pipes subject to saturation conditions are in metal contact with relatively warm water. The pipe surface in the lower condensation pipes is partly non-effective, due to the presence of liquid ammonia which has to pass through all the pipes to the bottom in order to get out, but this is not such an important factor as the lower pipe surface is not very much needed for liquefaction purposes. In all likelihood six pipes in the region of saturation are plenty and the 16, 18, 24 and 30 pipe condensers are uneconomical of pipe surface as far as *condensing* ammonia is concerned. The surface usually supplied is from 8 to 10 sq. ft. per ton of refrigeration.

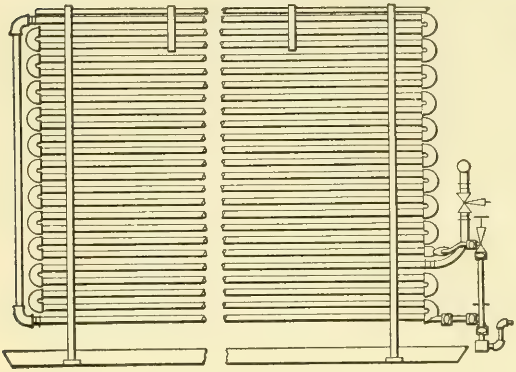


FIG. 90.—Common Atmospheric Ammonia Condenser.

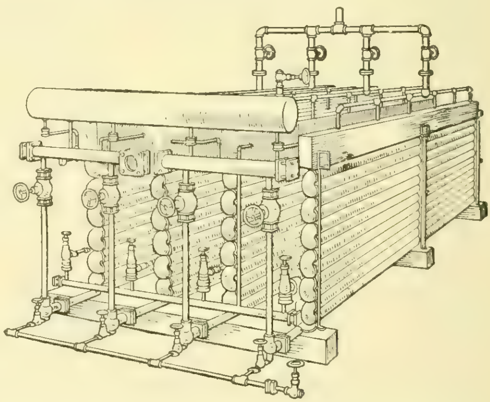


FIG. 91.—Four Stands of Atmospheric Condensers.

The Bleeder Condenser.—The bleeder type atmospheric condenser is

shown in Fig. 92. The hot gas always enters at the bottom and ascends so that the upper pipe is, to all intents and purposes, a dead end. The coldest water is showered over the entire length of the upper pipe and

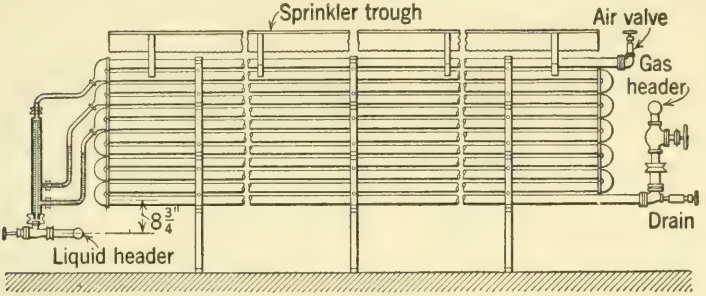


FIG. 92.—The Bleeder Type Condenser.

this is where the greatest load on the condenser occurs. Two and sometimes three tap offs are provided for the removal of the liquid. Tests indicate little difference in the static pressure between the gas entering the bottom and the gas in the top pipe, and this slight difference is

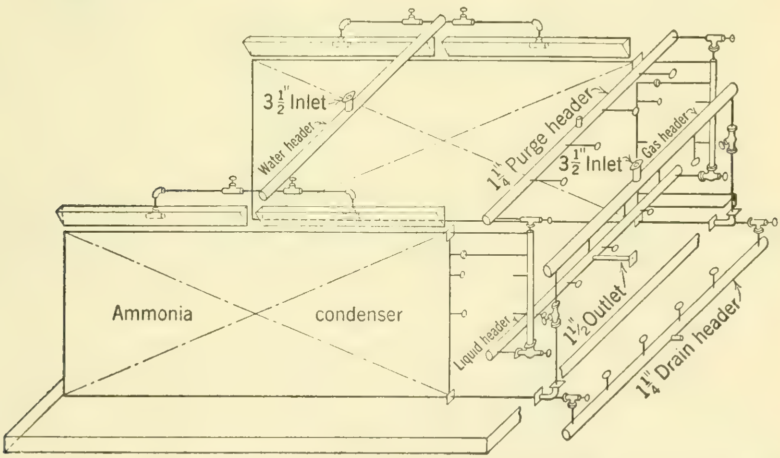


FIG. 93.—Six Stands, Atmospheric Ammonia Condensers, with Headers and Water Distributing Troughs.

easily accounted for in the resistance to the flow of the gas through the pipes and fittings. From 6 to 10 sq. ft. of surface are provided per ton of refrigeration as a rule.

The Flooded Atmospheric Condenser.—The flooded atmospheric condenser, (Fig. 94) uses an ejector nozzle, as shown in the insert, in order to

pick up the liquid condensate required for its proper performance. The lower pipes are practically similar in effect to those shown in the common atmospheric condenser. It would appear that, after the liquid is picked up by the gas, and is carried in more or less large plugs up to the upper pipe, on arrival at the upper part of the condenser the liquid separates from the gas and the two flow independent of one another except for the contact at the top of the liquid surface. The velocity of the gas in 2-in. pipe is small, being at usual (and standard) operating conditions about 8 ft. per second average velocity for the cross section of the pipe. With the flooded condenser the net area available for the passage of the gas would be one-half or possibly one-quarter of this amount and the velocity of the gas would be increased in proportion to the decrease of cross-sectional area. Remarkable performances have been obtained

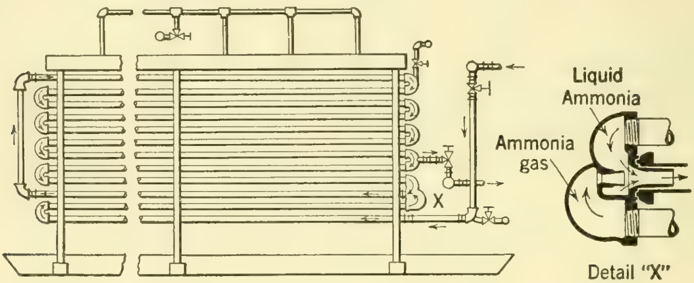


FIG. 94.—The Flooded Type Ammonia Condenser.

at times with this condenser, but when placed in parallel with other flooded condensers it is next to impossible to keep all the condensers performing equally. This condenser, also, is not one that will give the minimum head pressure, and this pressure may increase 20 lb. without any apparent cause.

The Double-pipe Condenser.—The atmospheric condenser needs to be used on the roof or otherwise located so as to expose freely the wetted surfaces to the air, which should pass through the stands freely. Where convenience or necessity requires the condenser to be located in the compressor room some other form or design must be used. For such conditions as in theatres, office buildings, hotels, etc., the double-pipe condenser needs to be used and (up to within the last few years) also for small installations.

The double-pipe condenser (Fig. 95), takes the gas at the top and the water enters at the bottom. The action of this condenser is counterflow, the refrigerant being inside the outer and the water inside the inner pipe. The water is therefore restrained, passing definitely from

pipe to pipe and has the number of passes corresponding to the number of pipes high of the stand. Having a fixed path the water velocity can be varied at will, something which is not true of any of the atmospheric condensers or of the vertical shell and tube condenser. The

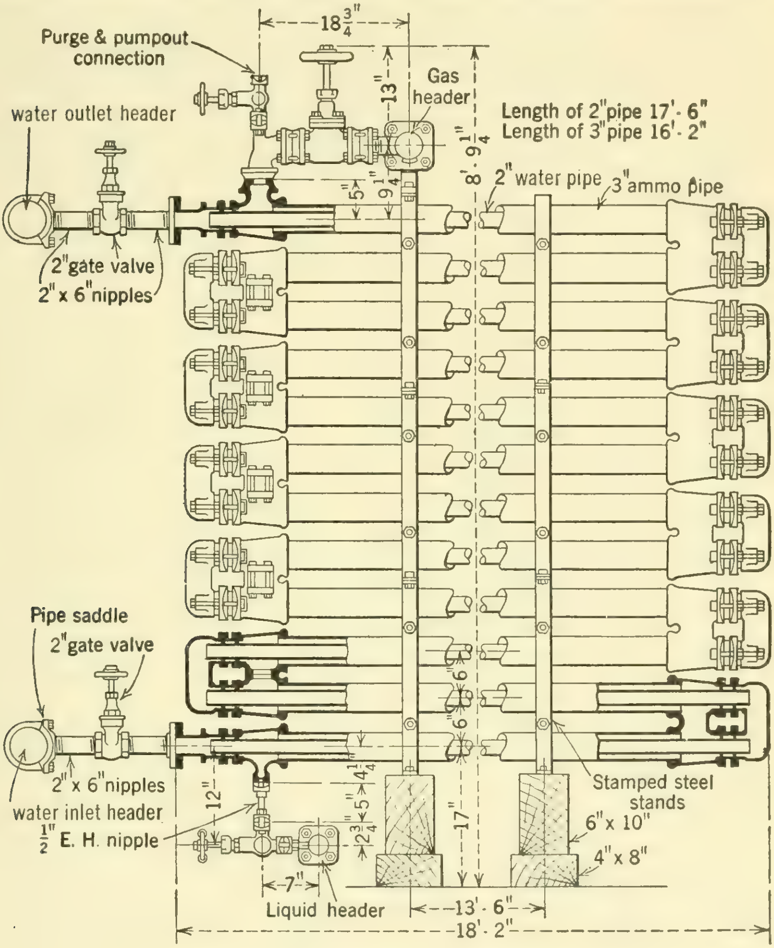


FIG. 95.—The Double-pipe Condenser.

capacity of the condenser can then be varied at will by changing the amount of water pumped through the condenser, although—incidentally—the friction head to be overcome by the water circulating pump increases very decidedly with an increase of the water velocity. The small cross-sectional area for the passage of the gas on entering the condenser causes some shock and vibration of the condenser and this

needs to be guarded against. There are also a large number of special fittings and special gaskets so that the condenser may have trouble from gas leaks unless the pipe and fittings are welded throughout, a resort which is becoming common in the case of the carbonic condenser.

As the gas enters the top and passes downward through the stand liquefaction takes place, but by far the greatest amount of the condensate takes place in the lower two pipes incidental with the coldest water. However, the sub-cooling effect due to the counterflow does not appear to be borne out in practice unless resistance to free draining or some other cause makes the liquid accumulate in the lower pipes. Sub-cooling is therefore not an inherent advantage of the double-pipe condenser, but indicates faulty piping or something not as it should be in a correctly designed and erected apparatus. This condenser is frequently a favorite for it has a large value of heat transfer, and is excellent in its performance, provided the water is reasonably free from scale-forming impurities. Unlike the atmospheric and the shell and tube condenser there is no appreciable splash of the water off of the condenser surfaces, which is a factor of extreme importance at times. The double-pipe condenser usually has from 6 to 10 sq. ft. per ton of refrigeration.

The Shell and Tube Condenser.—The shell and tube condenser (Fig. 96) is the latest and, from the design point of view, the best ammonia condenser built at the present time. Except for the small self-contained units, where the condenser is multi-pass and is placed horizontally, the shell and tube condenser is vertical and is designed for 2-in. tubes. The length of the tubes may vary from 6 ft.-0 in. in the 5- or 6-ton size to 10, 12 or 16 ft., depending in the larger sizes on the head room in the plant and the desires of the purchaser. The tubes are rolled into the tube sheet in a manner similar to a boiler tube. The tubes are laid out on approximately 2⅝-in. centers, with the rows of tubes at 60 degrees with one another similar to steam surface condenser practice. Open lanes can be provided and the gas enters in the center of the shell as a rule. As the condenser applied to refrigeration is comparatively new it is not clear what will be the outcome, but there is every indication that it will be the most popular design, especially in the larger sizes. It is compact to a remarkable degree as a 100-ton condenser needs to be only about 46 in. in diameter and 14 ft. to 2 in. total height and it is free from the multiplicity of valves and fittings, although there is a chance for the tubes to require additional rolling to maintain tightness.

The water distribution is obtained by means of saw-tooth distributors, by a distribution designed to give the condensing water a whirling action or other, the objective being to keep the water on the inside surface of the tubes and yet limit the amount of the thickness of the water to a

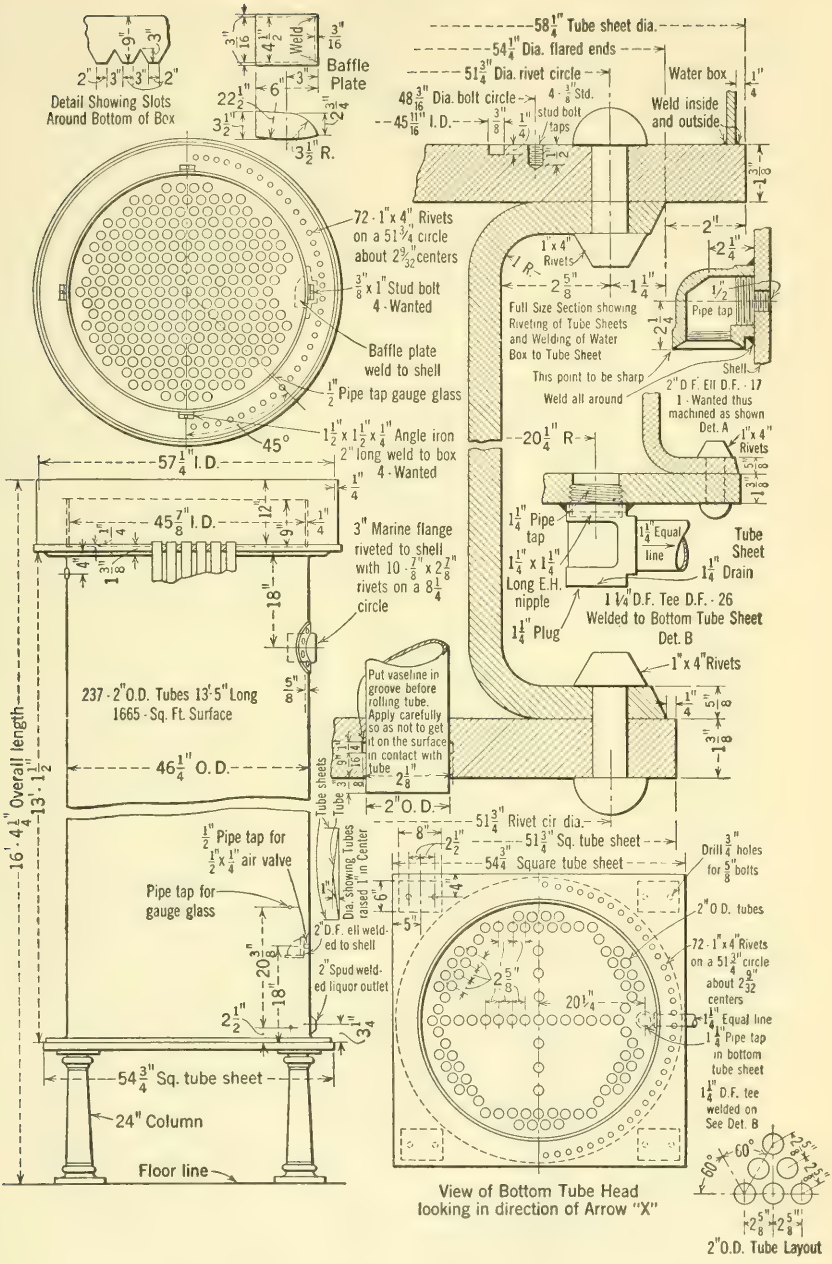


Fig. 96.—The Shell and Tube Condenser.

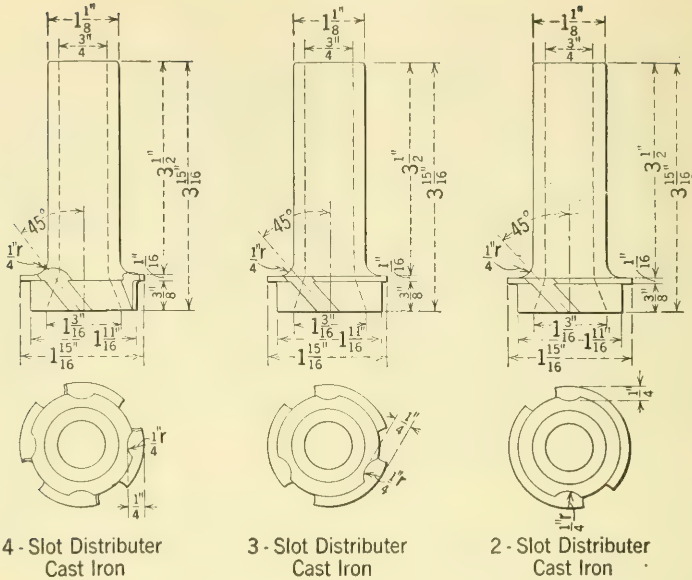


FIG. 97.—Water Distributors for Shell and Tube Condensers.

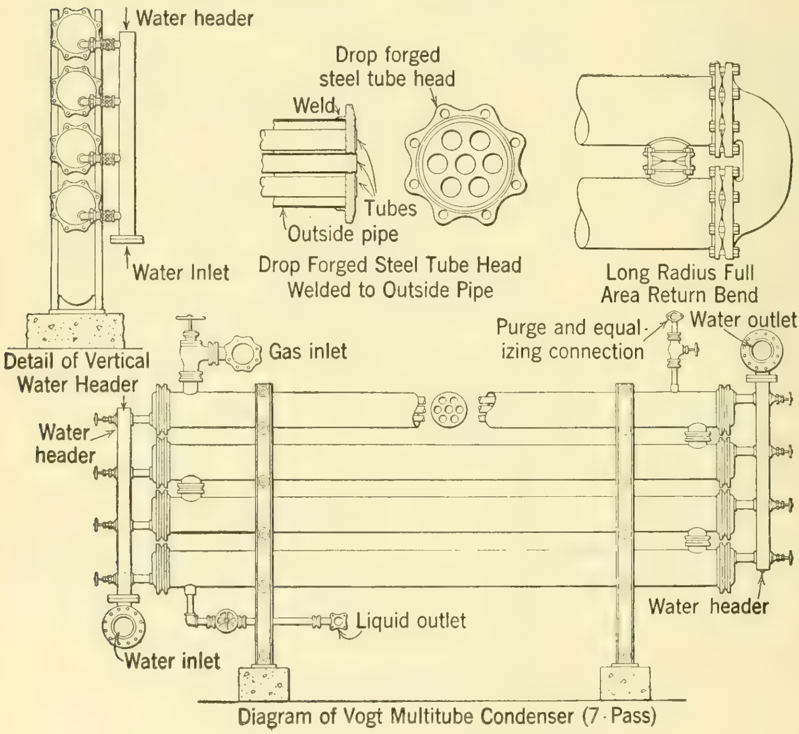


FIG. 98.—The Multitube Ammonia Condenser.

thin film. As the water enters the tubes at the top the greater part of the liquefaction takes place in the upper part of the condenser and the condensate flows by gravity down the tubes. There are, therefore, two liquid films—of liquid ammonia on the outside and of water on the inside of the tubes. However there is little tendency for the formation of an inert film on the surfaces and in consequence the rate of heat transfer is large and there is a prompt freeing of the surface of the liquid condensate and the heat transfer surface remains effective. The gas from the compressor may enter lanes arranged for it in the condenser so that the gas shock is negligible. Arranged in batteries there is no difficulty in making each condenser take its proper share of the load. A safety valve on the condenser will safeguard against excess pressure, which otherwise might cause rupture of the shell. The shell and tube condenser is designed at present (1927) for from 12 to 15 sq. ft. of heat transfer surface per ton of refrigeration, but tests by the author indicate that this amount is excessive.

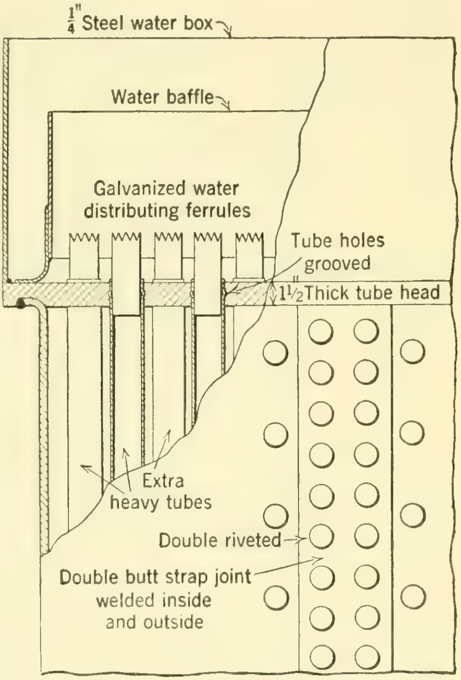


FIG. 99.—The Shell and Tube Condenser.

Heat Removed by Condenser.—The heat removed by the condenser is equal to the refrigerating effect plus the heat equivalent of the work done on the gas during the compression. This may be expressed *per ton of refrigeration per minute* as follows:

Q_c = the indicated horse power times 42.44 plus 200

$$= \frac{4.713}{E_r}\left[\frac{i_4 - i_3}{i_3 - i_2}\right] \times 42.44 + 200 = 200\left[\left(\frac{i_4 - i_3}{i_3 - i_2} \times \frac{1}{E_r}\right) + 1\right]$$

Where $i_4 - i_3$ is equal to the increase of the total heat per pound of ammonia during the compression, $i_3 - i_2$ is the refrigerating effect per pound of ammonia and E_r is the real volumetric efficiency of the compressor. With an expansion valve $i_1 = i_2$ (Fig. 1).

Water Requirements of The Condenser.—Amount of condenser water in gallons per ton per minute

$$= \frac{Q_c}{8.33 \times (t_2 - t_1)} = \frac{200\left(\frac{i_4 - i_3}{i_3 - i_1}\right) \times \left(\frac{1}{E_r} + 1\right)}{8.33 \times (t_2 - t_1)}$$

where the term $(t_2 - t_1)$ represents the rise in temperature of the condensing water.

Heat removed theoretically from different parts of the condenser per ton per minute.

a. $$Q_{\text{latent}} = \frac{200 \times r}{(i_3 - i_7)}$$

where

$$r = i_6 - i_1 \text{ (Fig 1)}$$

The points *1* and *7* will coincide, unless a separate aftercooler is used, if no inert gases are present in the condenser.

b. $$Q_{\text{aftercooler}} = \frac{200 \times (i_1 - i_7)}{(i_3 - i_7)}$$

c. $$Q_{\text{superheat}} = \frac{200 \times (i_4 - i_6)}{(i_3 - i_7)}$$

and the *average mean temperature* difference is:

$$t_m = \frac{Q_l\, t_l + Q_a\, t_a + Q_s\, t_s}{Q_c}$$

where t_a, t_b, and t_c are the average temperature differences in the latent heat, the aftercooler and the superheat regions respectively.

The Effect of Inert Gases in the Condenser.—In steam condenser practice using the surface condenser it has been found that the presence of *air* in the steam has the effect of reducing very greatly the coefficient of heat transfer and that this value could be from 5 to 10 times as great as is found in actual practice if air could be eliminated. The effect of this air in the surface condenser is to mix with the condensible gases, and as condensation continues, to become more and more concentrated as it approaches the surface of the tubes. The result of this is the development of a non-condensible gas film on the metal surface. As the amount of air in the condenser increases the thickness of this air film increases, resulting in a lowering of the value of the coefficient of heat transfer. In addition to the loss of surface efficiency the total pressure of the condenser is increased in accordance with Dalton's law which says that the total pressure is the sum of the partial pressures,

and that the partial pressures exerted by each gas or vapor are those which would be caused by each occupying the volume by themselves. In other words,

$$p = p_1 - p_2$$

where p_1 = partial pressure of the air

p_2 = partial pressure of (in this case) the ammonia

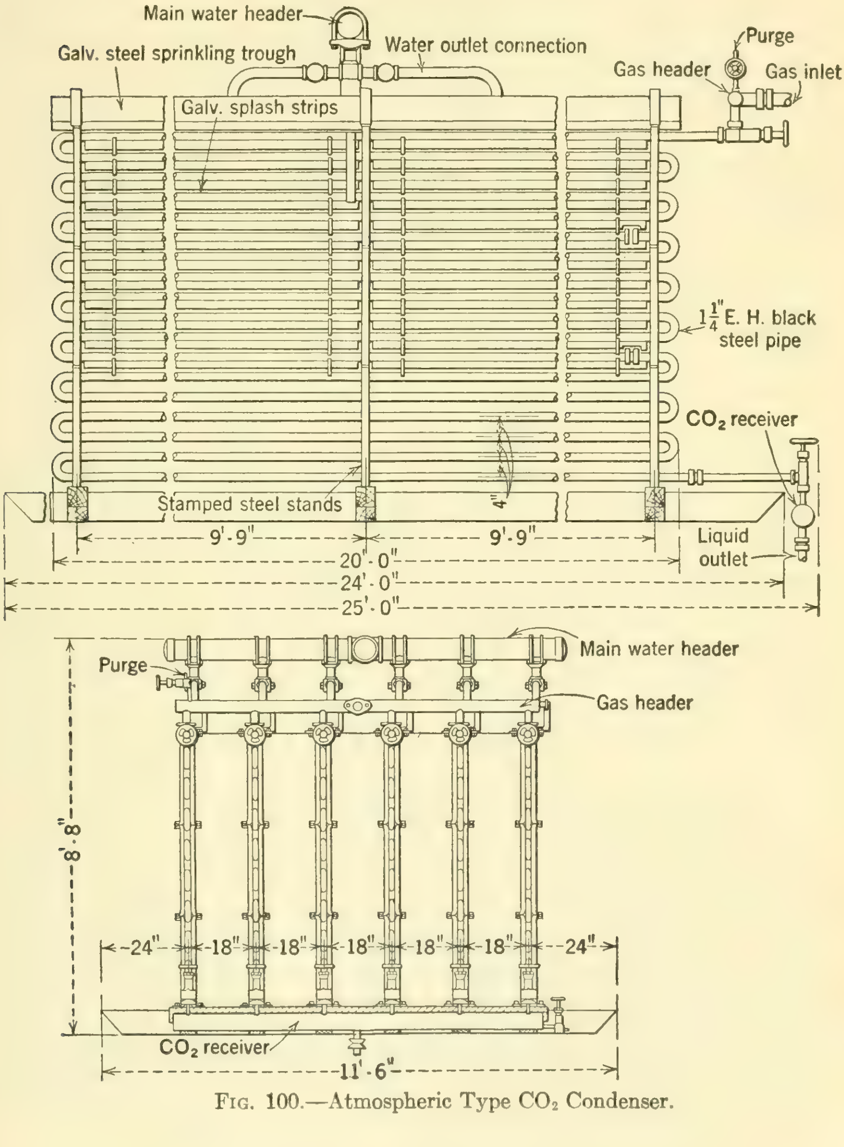


FIG. 100.—Atmospheric Type CO_2 Condenser.

In the case of the ammonia condenser the partial pressure of the air in pounds per square inch could be calculated, if the volume of the condenser were known and the temperature and the weight of the air present could be determined. Then by means of the characteristic equation for a perfect gas (air),

$$pv = MBT$$

where M = weight of gas in pounds;
B = gas constant = 53.3 for air;
T = absolute temperature of the air in degrees F.;
v = volume in cubic feet of the gas under a pressure p lb. per square inch absolute.

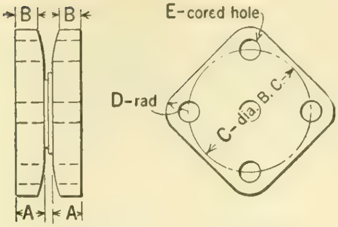
The partial pressure of the ammonia is that corresponding to the temperature during liquefaction, as given in the ammonia tables.

For example a stand of 12–2 in. pipes, 20 ft. long, has a volume of approximately $\frac{12 \times 20}{42.9}$ cu. ft. and the weight of air that would exert a pressure of 10 lb. per square inch at a temperature of 70 deg. F. is

$$M = \frac{pv}{BT} = \frac{10 \times 12 \times 20 \times 144}{42.9 \times 53.3 \times 530} = 0.285 \text{ lb.}$$

As the refrigerating system is a closed one and the pressure is normally always greater than atmospheric pressure there is not much reason for air to get into the condenser. However there is always some air remaining in the system on charging, or whenever repairs are made to any part of the plant and undoubtedly some air enters the compressor when it is operated at a suction pressure below that of the atmosphere. The action of the boiling of the refrigerant in the evaporating coils, and the velocity of the gas flow into the compressor and the condenser results in the picking up of the air and the final depositing of it in the condenser. Any non-condensible gas which enters the condenser will stay there until it is pumped out or leaks out (with the ammonia) and as it accumulates the pressure will increase.

The old method of purging non-condensible gases from the condenser was to open the purge valve and permit some of the gas to escape to the atmosphere. However to save ammonia some special device is necessary, for if air once gets into the condenser it is next to impossible to get rid of it, by purging, without loss of ammonia. The best method of purging is to permit a slight flow of gas from the condenser, or from the top of the liquid receiver, either continually or intermittently and to refrigerate this gas by means of a special evaporating coil. The gas permitted to escape under these conditions will be air in most part,



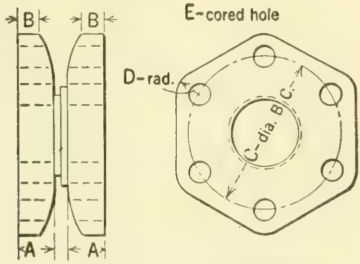
CO^2 Flange - 4-Bolt Type

a

CO$_2$ Flange
4-Bolt Type

Size	Symbol Number	A	B	C	D	E	Size of Bolts
$\frac{3}{4}$	1024-B	$\frac{7}{8}$	$\frac{5}{8}$	$3\frac{1}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{1}{2} \times 2\frac{7}{8}$
1	1024-C	$1\frac{1}{16}$	$\frac{3}{4}$	$3\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{5}{8} \times 3\frac{3}{8}$
$1\frac{1}{4}$	1024-D	$1\frac{1}{8}$	$\frac{3}{4}$	$4\frac{1}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{5}{8} \times 3\frac{1}{2}$
$1\frac{1}{2}$	1024-E	$1\frac{1}{8}$	$\frac{3}{4}$	$4\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{5}{8} \times 3\frac{1}{2}$
2	1024-F	$1\frac{1}{4}$	$\frac{15}{16}$	$5\frac{1}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{3}{4} \times 3\frac{7}{8}$

All dimensions given in inches unless otherwise specified.



CO^2 Flange - 6-Bolt Type

b

CO$_2$ Flange
6-Bolt Type

Size	Symbol Number	A	B	C	D	E	Size of Bolts
$2\frac{1}{2}$	1292-F	$1\frac{5}{8}$	1	$6\frac{1}{2}$	1	$\frac{7}{8}$	$\frac{3}{4} \times 5\frac{1}{4}$
3	1293-F	$1\frac{3}{4}$	$1\frac{1}{8}$	$7\frac{3}{4}$	1	1	$\frac{7}{8} \times 5$
4	1294-F	$2\frac{1}{4}$	$1\frac{1}{4}$	$9\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{8} \times 6$

All dimensions given in inches unless otherwise specified.

Fig. 101.—Carbonic Fittings—Flanges.

although the volume of gas in the holder contains as much ammonia as would be present at that temperature were there no air present, and the weight of ammonia purged would be the total volume of gas purged (at the pressure of the tank) divided by specific volume of ammonia at the temperature maintained by the cooling coils. If one cubic foot is purged and the temperature is 0 deg. F. then $\frac{1}{9.12} = 0.11$ lb. of ammonia is lost, but if the temperature is 32 deg. F. then $\frac{1}{4.64} = 0.216$ lb. will be the amount of ammonia lost.

TABLE 19

RECEIVERS—HIGH-PRESSURE OIL SEPARATORS—AMMONIA ACCUMULATORS

Ammonia Receivers

Tons Ice Making	Diameter, Inches	Length, Feet	Length, Inches	Pipe Connections, Inches	Weight, Pounds
1 and 2	12	4	0	½	290
3	12	6	0	½	375
4	12	8	0	½	460
5 and 6	16	7	0	½	475
8 to 20	20	7	0	¾	650
25	20	10	0	1	850
30 to 50	20	16	0	1¼	1200
60 to 100	24	14	0	2	1600
125 to 200	24	16	0	2	1770

H. P. Oil Separators

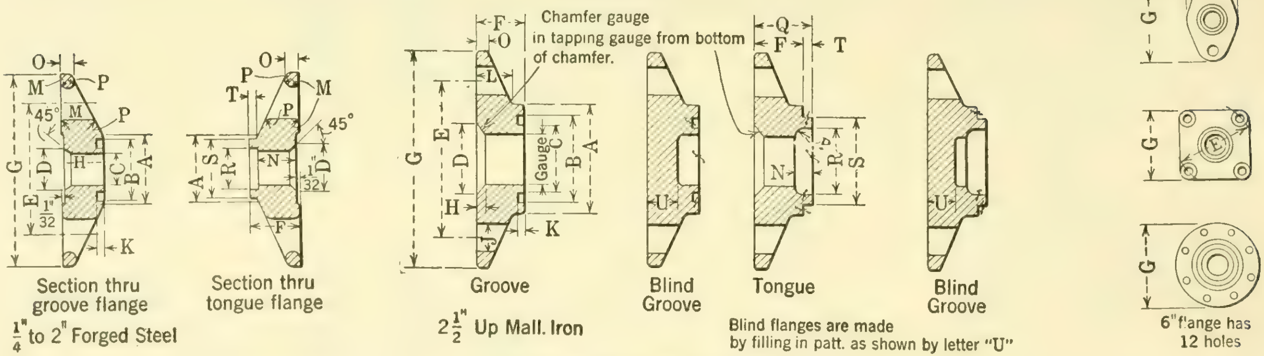
Tons Ice Making	Diameter, Inches	Length, Feet	Length, Inches	Pipe Connections, Inches	Weight, Pounds
1 to 12	12	3	0	1¼ and 2	220
15 to 35	16	4	0	2½ and 3	370
40 to 60	20	5	0	3½ and 4	600
75 to 100	24	5	0	5 and 6	890

Ammonia Accumulators

Tons Ice Making	Diameter, Inches	Length, Feet	Length, Inches	Pipe Connections, Inches	Weight, Pounds
1 to 12	12	3	0	1¼ and 2	285
15 to 25	16	4	0	2½ and 3	400
30 to 75	20	5	0	3½ and 4	650
100 to 200	24	5	0	5 and 6	910

TABLE 20

Dimensions of Ammonia Companion Flanges. Tongue and Groove—Straight Backs
(Crane Co.)



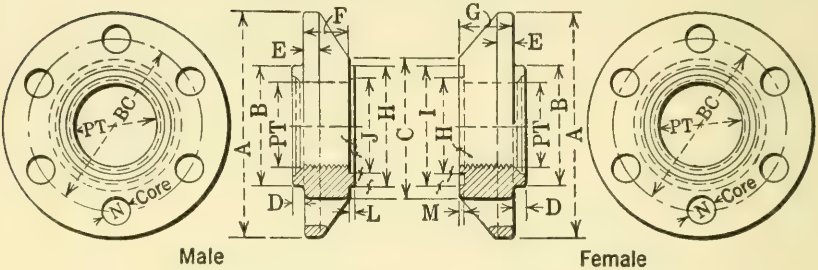
All Dimensions Given in Inches

	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	Q	R	S	T	U	V
1/4	1 1/8	1	9/16	21/32	2 3/8	7/8	3 5/8	21/32	5/8	1/8	......	1/8	21/32	9/32	5/32	......	19/32	31/32	3/16	......	1 5/8
3/8	1 1/4	1 1/8	11/16	13/16	2 9/16	31/32	3 13/16	3/4	5/8	1/8	......	1/8	3/4	9/32	5/32	......	23/32	1 3/32	3/16	......	1 3/4
1/2	1 3/8	1 1/4	27/32	31/32	2 3/4	1 1/16	4	27/32	5/8	1/8	......	1/8	27/32	9/32	5/32	......	7/8	1 7/32	3/16	......	2
3/4	1 5/8	1 1/2	1 3/32	1 7/32	3 1/4	1 3/32	4 3/4	29/32	3/4	1/8	......	1/8	29/32	9/32	5/32	......	1 1/8	1 15/32	3/16	......	2 1/4
1	2	1 13/16	1 11/32	1 15/32	3 1/4	1 7/32	3 9/16	1	5/8	1/8	......	5/32	1	9/32	......	......	1 3/8	1 25/32	3/16		
1 1/4	2 7/16	2 1/4	1 21/32	1 13/16	3 3/4	1 11/32	3 15/16	1 1/16	5/8	3/16	......	5/32	1 1/16	5/16	......	......	1 11/16	2 7/32	1/4	......	......
1 1/2	2 11/16	2 1/2	1 29/32	2 3/32	4 1/2	1 13/32	4 11/16	1 1/8	3/4	3/16	......	5/32	1 1/8	11/32	......	......	1 15/16	2 15/32	1/4		
2	3 5/16	3 1/8	2 13/32	2 19/32	5	1 7/16	5 1/16	1 5/32	3/4	3/16	......	3/16	1 5/32	3/8	......	......	2 7/16	3 3/32	1/4		
2 1/2	4 1/4	3 11/16	2 29/32	3 5/64	5 7/8	1 7/16	5 15/16	1/4	7/8	1/4	1 1/4	3/16	21/32	3/8	9/32	1 7/8	2 15/16	3 21/32	7/16	1	
3	4 7/8	4 5/16	3 17/32	3 3/4	6 5/8	1 9/16	8 1/4	17/64	7/8	1/4	1 3/8	3/16	45/64	3/8	9/32	2	3 9/16	4 9/32	7/16	1	
3 1/2	5 1/2	4 7/8	4 1/32	4 1/4	7 1/4	1 5/8	9	19/64	7/8	1/4	1 7/16	3/16	43/64	3/8	9/32	2 1/16	4 1/16	4 27/32	7/16	1 1/16	
4	6 1/8	5 1/2	4 17/32	4 25/32	7 7/8	1 11/16	10	5/16	7/8	1/4	1 1/2	3/16	21/32	7/16	5/16	2 1/8	4 9/16	5 15/32	7/16	1 1/8	
5	7 5/8	6 7/8	5 19/32	5 7/8	9 1/4	1 27/32	11	19/64	7/8	1/4	1 5/8	3/16	47/64	1/2	5/16	2 9/32	5 5/8	6 27/32	7/16	1 3/16	
6	8 3/4	7 7/8	6 21/32	7	10 5/8	2	12 1/2	3/8	7/8	1/4	1 11/16	3/16	3/4	1/2	5/16	2 7/16	6 11/16	7 27/32	7/16	1 1/4	

TABLE 21

Dimensions of Ammonia Companion Flanges

Tongue and Groove—Crane Co.

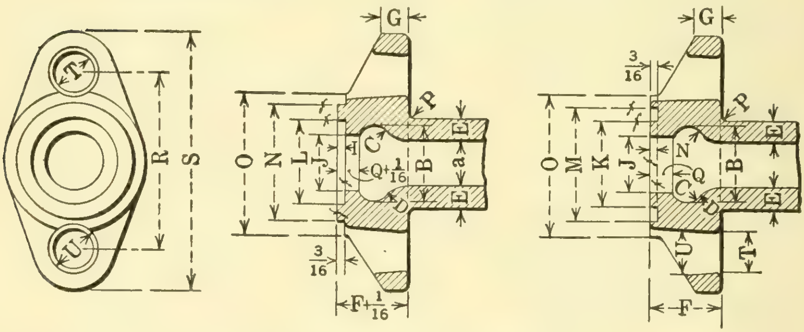


All Dimensions Given in Inches

	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P
7	14	$9\frac{1}{2}$	$8\frac{5}{8}$	$9\frac{5}{8}$	$9\frac{11}{16}$	$8\frac{9}{16}$	$2\frac{1}{16}$	1.61	$1\frac{13}{16}$	$1\frac{1}{2}$	$7\frac{7}{8}$	$\frac{1}{4}$	$\frac{3}{16}$	$11\frac{7}{8}$	12– $\frac{7}{8}$
8	15	$10\frac{5}{8}$	$9\frac{5}{8}$	$10\frac{7}{8}$	$10\frac{15}{16}$	$9\frac{9}{16}$	$2\frac{3}{16}$	1.71	$1\frac{29}{32}$	$1\frac{5}{8}$	$8\frac{7}{8}$	$\frac{1}{4}$	$\frac{3}{16}$	13	12– $\frac{7}{8}$
10	$17\frac{1}{2}$	$12\frac{5}{8}$	$11\frac{5}{8}$	$13\frac{1}{8}$	$13\frac{3}{16}$	$11\frac{9}{16}$	$2\frac{3}{8}$	1.92	$2\frac{7}{64}$	$1\frac{7}{8}$	11	$\frac{1}{4}$	$\frac{3}{16}$	$15\frac{1}{4}$	16–1
12	$20\frac{1}{2}$	$14\frac{3}{4}$	$13\frac{5}{8}$	$15\frac{1}{8}$	$15\frac{3}{16}$	$13\frac{9}{16}$	$2\frac{9}{16}$	2.12	$2\frac{5}{16}$	2	13	$\frac{1}{4}$	$\frac{3}{16}$	$17\frac{3}{4}$	16–$1\frac{1}{8}$

TABLE 22

Flange Dimensions of "York" Oval Flange Fittings

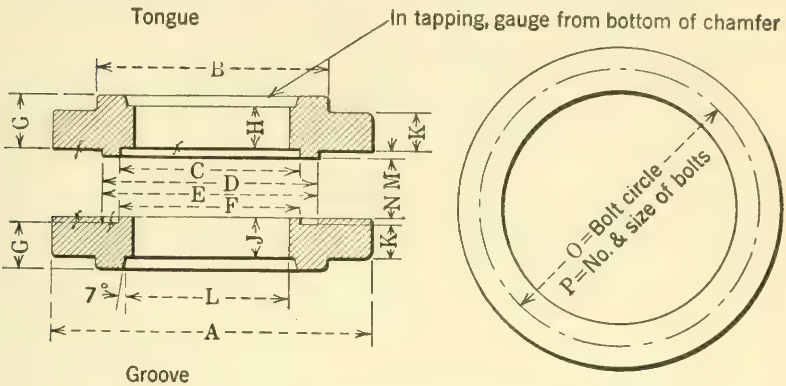


All Dimensions Given in Inches

Size	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U
$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{5}{16}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{15}{32}$	$\frac{7}{8}$	$\frac{29}{32}$	$1\frac{3}{8}$	$1\frac{11}{32}$	$1\frac{3}{4}$	$\frac{1}{16}$	$\frac{3}{8}$	$2\frac{1}{4}$	$3\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$
$\frac{3}{8}$ and $\frac{1}{2}$	$\frac{5}{8}$	$\frac{15}{16}$	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{5}{16}$	1	$\frac{5}{16}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{11}{16}$	$1\frac{1}{4}$	$1\frac{9}{32}$	$1\frac{3}{4}$	$1\frac{23}{32}$	$2\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{8}$	$2\frac{5}{8}$	$3\frac{3}{4}$	$\frac{9}{16}$	$\frac{5}{8}$
$\frac{3}{4}$	$\frac{3}{4}$	$1\frac{3}{16}$	$\frac{5}{16}$	$\frac{1}{2}$	$\frac{5}{16}$	$1\frac{1}{8}$	$\frac{3}{8}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{15}{16}$	$1\frac{5}{16}$	$1\frac{11}{32}$	$1\frac{13}{16}$	$1\frac{25}{32}$	$2\frac{3}{16}$	$\frac{1}{16}$	$\frac{1}{2}$	$2\frac{3}{4}$	$4\frac{1}{8}$	$\frac{11}{16}$	$\frac{3}{4}$
1	1	$1\frac{7}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{16}$	$1\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{8}$	$\frac{5}{32}$	$1\frac{3}{16}$	$1\frac{5}{8}$	$1\frac{21}{32}$	$2\frac{1}{8}$	$2\frac{3}{32}$	$2\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{2}$	$3\frac{1}{8}$	$4\frac{1}{2}$	$\frac{11}{16}$	$\frac{3}{4}$
$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{3}{4}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{16}$	$1\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{8}$	$\frac{5}{32}$	$1\frac{15}{32}$	$1\frac{13}{16}$	$1\frac{27}{32}$	$2\frac{5}{16}$	$2\frac{9}{32}$	$2\frac{5}{8}$	$\frac{1}{16}$	$\frac{1}{2}$	$3\frac{5}{16}$	$4\frac{7}{8}$	$\frac{11}{16}$	$\frac{3}{4}$

TABLE 23

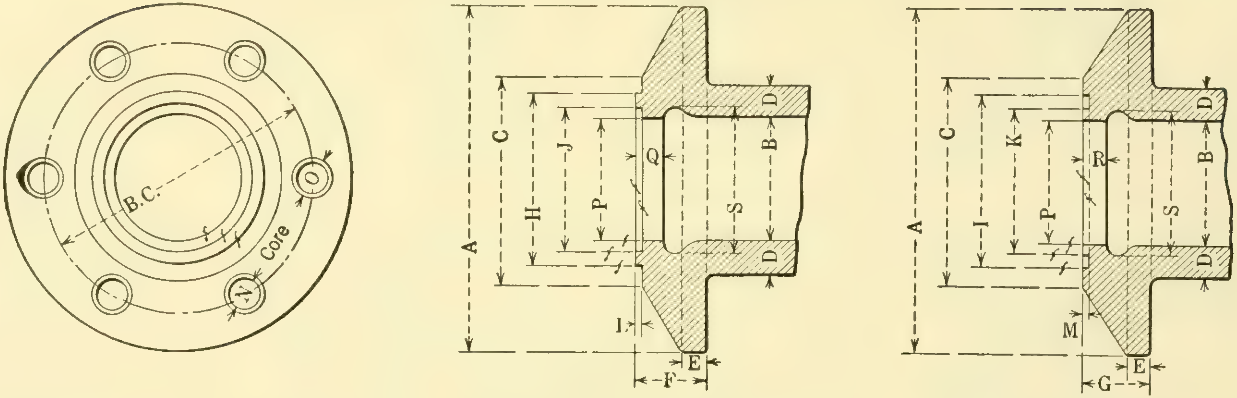
Dimensions of "York" Tongue and Groove Round Flanges



All Dimensions Given in Inches

Size, P.T.	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*J*	*K*	*L*	*M*	*N*, Core	Bolts		
															B.C.	No.	Size
$2\frac{1}{2}$	$7\frac{1}{8}$	$3\frac{5}{8}$	$4\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$1\frac{7}{16}$	$1\frac{5}{8}$	$3\frac{3}{4}$	$3\frac{25}{32}$	3	$2\frac{31}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	$\frac{7}{8}$	$5\frac{1}{2}$	6	$\frac{3}{4}\times 4$
3	$7\frac{7}{8}$	$4\frac{5}{16}$	$4\frac{3}{4}$	$\frac{3}{8}$	$\frac{9}{16}$	$1\frac{9}{16}$	$1\frac{3}{4}$	$4\frac{3}{8}$	$4\frac{13}{32}$	$3\frac{5}{8}$	$3\frac{19}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	$\frac{7}{8}$	$6\frac{1}{8}$	6	$\frac{3}{4}\times 4\frac{1}{4}$
$3\frac{1}{2}$	$8\frac{5}{8}$	$4\frac{7}{8}$	$5\frac{3}{8}$	$\frac{3}{8}$	$\frac{9}{16}$	$1\frac{5}{8}$	$1\frac{13}{16}$	$4\frac{7}{8}$	$4\frac{29}{32}$	$4\frac{1}{8}$	$4\frac{3}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	$\frac{7}{8}$	$6\frac{7}{8}$	8	$\frac{3}{4}\times 4\frac{1}{2}$
4	$9\frac{3}{8}$	$5\frac{3}{8}$	$6\frac{1}{8}$	$\frac{3}{8}$	$\frac{5}{8}$	$1\frac{11}{16}$	$1\frac{7}{8}$	$5\frac{1}{2}$	$5\frac{17}{32}$	$4\frac{3}{4}$	$4\frac{23}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	$\frac{7}{8}$	$7\frac{5}{8}$	8	$\frac{3}{4}\times 4\frac{1}{2}$
$4\frac{1}{2}$	$9\frac{3}{4}$	$5\frac{13}{16}$	$6\frac{1}{2}$	$\frac{3}{8}$	$\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{15}{16}$	6	$6\frac{1}{32}$	$5\frac{1}{4}$	$5\frac{7}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	$\frac{7}{8}$	8	8	$\frac{3}{4}\times 4\frac{3}{4}$
5	$10\frac{1}{4}$	$6\frac{7}{16}$	7	$\frac{3}{8}$	$\frac{11}{16}$	$1\frac{13}{16}$	2	$6\frac{1}{2}$	$6\frac{17}{32}$	$5\frac{3}{4}$	$5\frac{23}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	$\frac{7}{8}$	$8\frac{1}{2}$	8	$\frac{3}{4}\times 4\frac{3}{4}$
6	$11\frac{1}{2}$	$7\frac{5}{8}$	$8\frac{1}{4}$	$\frac{7}{16}$	$\frac{3}{4}$	$1\frac{7}{8}$	$2\frac{1}{16}$	$7\frac{3}{4}$	$7\frac{25}{32}$	7	$6\frac{31}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	$\frac{7}{8}$	$9\frac{3}{4}$	8	$\frac{3}{4}\times 5$
7	$12\frac{1}{2}$	$8\frac{5}{8}$	9	$\frac{7}{16}$	$\frac{13}{16}$	$2\frac{1}{8}$	$2\frac{5}{16}$	$8\frac{5}{8}$	$8\frac{21}{32}$	$7\frac{7}{8}$	$7\frac{27}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	1	$10\frac{3}{4}$	8	$\frac{7}{8}\times 5\frac{1}{2}$
8	$14\frac{1}{4}$	$9\frac{3}{4}$	$10\frac{1}{4}$	$\frac{1}{2}$	$\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{1}{2}$	$9\frac{7}{8}$	$9\frac{29}{32}$	9	$8\frac{31}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	1	12	8	$\frac{7}{8}\times 6$
10	$16\frac{1}{2}$	12	$12\frac{1}{2}$	$\frac{5}{8}$	$\frac{15}{16}$	$2\frac{3}{8}$	$2\frac{5}{8}$	12	$12\frac{1}{32}$	11	$10\frac{31}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	$1\times 1\frac{1}{8}$	$14\frac{1}{4}$	12	$\frac{7}{8}\times 6\frac{1}{4}$
12	$18\frac{3}{4}$	$14\frac{1}{8}$	$14\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{16}$	$2\frac{1}{2}$	$2\frac{3}{4}$	$14\frac{1}{4}$	$14\frac{9}{32}$	$13\frac{1}{4}$	$13\frac{7}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	$1\frac{1}{4}\times 1\frac{1}{4}$	$16\frac{1}{2}$	12	$1\times 6\frac{1}{2}$

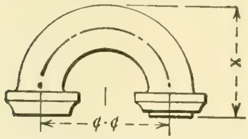
TABLE 24
Flange Dimensions of "York" Tongue and Groove Round Flange Fittings



All Dimensions Given in Inches

Size	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	1/16-in. Gasket	Bolts B.C.	Bolts No.	Bolts Fitt. & Fitt.	Bolts Fitt. & Flge.
2 1/2	7 1/8	2 5/8	4 1/4	5/8	1/2	1 7/16	1 3/8	3 3/4	3 25/32	3	2 31/32	5/32	1/8	13/16	7/8	2 9/16	9/16	1/2	3	3 × 3 3/4	5 1/2	6	3/4 × 3 3/4	3/4 × 4
3	7 7/8	3 1/8	4 3/4	5/8	9/16	1 9/16	1 1/2	4 3/8	4 13/32	3 5/8	3 19/32	5/32	1/8	13/16	7/8	3 1/8	9/16	1/2	3 5/8	3 5/8 × 4 3/8	6 1/8	6	3/4 × 4	3/4 × 4 1/4
3 1/2	8 5/8	3 5/8	5 3/8	11/16	9/16	1 9/16	1 1/2	4 7/8	4 29/32	4 1/8	4 3/32	5/32	1/8	13/16	7/8	3 11/16	9/16	1/2	4 1/8	4 1/8 × 4 7/8	6 7/8	8	3/4 × 4	3/4 × 4 1/4
4	9 3/8	4 1/8	6 1/8	11/16	5/8	1 11/16	1 5/8	5 1/2	5 17/32	4 3/4	4 23/32	5/32	1/8	13/16	7/8	4 3/16	9/16	1/2	4 3/4	4 3/4 × 5 1/2	7 5/8	8	3/4 × 4 1/4	3/4 × 4 1/2
4 1/2	9 3/4	4 5/8	6 1/2	3/4	5/8	1 3/4	1 11/16	6	6 1/32	5 1/4	5 7/32	5/32	1/8	13/16	7/8	4 5/8	9/16	1/2	5 1/4	5 1/4 × 6	8	8	3/4 × 4 1/2	3/4 × 4 3/4
5	10 1/4	5 1/8	7	3/4	11/16	1 13/16	1 3/4	6 1/2	6 17/32	5 3/4	5 23/32	5/32	3/4	13/16	7/8	5 1/4	9/16	1/2	5 7/8	5 3/4 × 6 1/2	8 1/2	8	3/4 × 4 1/2	3/4 × 4 3/4
6	11 1/2	6 1/8	8 1/4	3/4	3/4	1 15/16	1 7/8	7 3/4	7 25/32	7	6 31/32	5/32	3/4	13/16	7/8	6 1/8	9/16	1/2	6 7/8	7 × 7 3/4	9 3/4	8	3/4 × 4 3/4	3/4 × 5
7	12 1/2	7 1/8	9	3/4	13/16	2	1 15/16	8 5/8	8 21/32	7 7/8	7 27/32	5/32	1/8	15/16	1	7 1/8	9/16	1/2	8	7 7/8 × 8 5/8	10 3/4	8	7/8 × 5	7/8 × 5 1/4
8	14 1/4	8 1/8	10 1/4	13/16	7/8	2 1/16	2	9 7/8	9 29/32	9	8 31/32	5/32	1/8	15/16	1	8 1/8	9/16	1/2	9	9 × 9 7/8	12	8	7/8 × 5 1/4	7/8 × 5 3/4
10	16 1/2	10 1/8	12 1/2	1	15/16	2 1/16	2	12	12 1/32	11	10 31/32	5/32	1/8	15/16	1	10 1/8	9/16	1/2	11	11 × 12	14 1/4	12	7/8 × 5 1/4	7/8 × 5 3/4
12	18 3/4	12 1/8	14 3/4	1	1 1/16	2 3/16	2 1/8	14 1/4	14 9/32	13 1/4	13 7/32	5/32	1/8	1 1/8	1 1/4	12 1/8	9/16	1/2	13 1/8	13 1/4 × 14 1/4	16 1/2	12	1 × 5 1/2	1 × 6

TABLE 25
Length of "York" Return Bends
Flanged Bends



All Dimensions Given in Inches

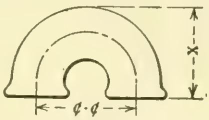
Oval Flanged

Size	¢–¢	x	Size	¢–¢	x	Size	¢–¢	x
$1\frac{1}{4}$	$3\frac{3}{8}$	$4\frac{1}{16}$	$1\frac{1}{4}$	$4\frac{1}{2}$	$4\frac{5}{8}$	$1\frac{1}{4}$	6	$5\frac{5}{8}$
$1\frac{1}{2}$	4	$4\frac{3}{8}$						

Square Flanged

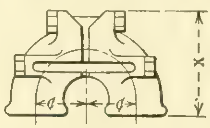
Size	¢–¢	x	Size	¢–¢	x	Size	¢–¢	x
$1\frac{1}{4}$	$4\frac{1}{8}$	$4\frac{5}{8}$	2	8	$6\frac{7}{8}$	$2\frac{1}{2}$	$5\frac{3}{4}$	$4\frac{3}{4}$
$1\frac{1}{4}$	$8\frac{1}{2}$	$6\frac{11}{16}$	2	$8\frac{1}{2}$	$7\frac{1}{8}$	3	12	10
$1\frac{1}{2}$	6	$5\frac{9}{16}$	2	9	$7\frac{3}{8}$	4	12	$10\frac{11}{16}$
$1\frac{1}{2}$	8	$6\frac{9}{16}$	2	10	$7\frac{7}{8}$	$1\frac{1}{4}$	4	$4\frac{1}{2}$
2	$4\frac{5}{8}$	$4\frac{5}{8}$	2	12	$8\frac{15}{16}$	$1\frac{1}{4}$	6	$5\frac{1}{4}$
2	6	$5\frac{7}{8}$	2	15	$5\frac{5}{8}$			

Solid Screw Bends



Size	¢–¢	x	Size	¢–¢	x	Size	¢–¢	x
1	$1\frac{3}{4}$	$2\frac{9}{16}$	$1\frac{1}{4}$	7	$5\frac{9}{16}$	2	$3\frac{1}{2}$	$4\frac{1}{8}$
1	$2\frac{1}{2}$	$2\frac{15}{16}$	$1\frac{1}{4}$	8	$5\frac{15}{16}$	2	$3\frac{3}{4}$	$4\frac{3}{8}$
1	3	$3\frac{3}{16}$	$1\frac{1}{4}$	11	$7\frac{9}{16}$	2	4	$4\frac{1}{2}$
1	4	$3\frac{13}{16}$	$1\frac{1}{2}$	$2\frac{9}{32}$	$3\frac{1}{4}$	2	$4\frac{3}{4}$	$4\frac{7}{8}$
1	6	$4\frac{13}{16}$	$1\frac{1}{2}$	$2\frac{3}{4}$	$3\frac{11}{16}$	2	6	$5\frac{9}{16}$
$1\frac{1}{4}$	$2\frac{1}{4}$	$3\frac{5}{16}$	$1\frac{1}{2}$	$2\frac{3}{4}$	$4\frac{1}{16}$	2	8	$6\frac{1}{2}$
$1\frac{1}{4}$	$2\frac{1}{2}$	$3\frac{1}{4}$	$1\frac{1}{2}$	3	$3\frac{13}{16}$	2	10	$7\frac{1}{2}$
$1\frac{1}{4}$	3	$3\frac{3}{8}$	$1\frac{1}{2}$	4	$4\frac{3}{16}$	$2\frac{1}{2}$	6	$6\frac{1}{16}$
$1\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{11}{16}$	$1\frac{1}{2}$	6	$5\frac{3}{16}$	3	6	$6\frac{3}{8}$
$1\frac{1}{4}$	4	$3\frac{15}{16}$	$1\frac{1}{2}$	8	$6\frac{3}{16}$	$3\frac{1}{2}$	6	$6\frac{5}{8}$
$1\frac{1}{4}$	$4\frac{1}{2}$	$4\frac{7}{16}$	2	3	$4\frac{5}{8}$	4	8	$8\frac{1}{4}$
$1\frac{1}{4}$	5	$4\frac{11}{16}$	2	3	$3\frac{15}{16}$	4	10	$9\frac{5}{16}$
$1\frac{1}{4}$	6	$4\frac{15}{16}$	2	$3\frac{1}{4}$	$4\frac{1}{16}$	4	12	$10\frac{1}{4}$
$1\frac{1}{4}$	$6\frac{3}{4}$	$5\frac{9}{16}$						

Split Screw Bends

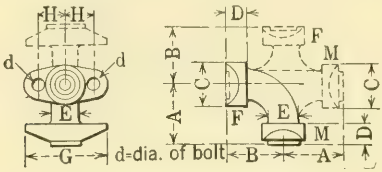


Size	¢–¢	x	Size	¢–¢	x	Size	¢–¢	x
1	$2\frac{1}{2}$	3	$1\frac{1}{4}$	5	$5\frac{5}{8}$	2	4	$5\frac{9}{16}$
1	3	$3\frac{1}{4}$	$1\frac{1}{4}$	6	$6\frac{1}{8}$	2	$4\frac{3}{4}$	$5\frac{15}{16}$
1	4	$3\frac{3}{4}$	$1\frac{1}{2}$	4	$5\frac{1}{8}$	2	6	$6\frac{9}{16}$
1	6	$4\frac{3}{4}$	$1\frac{1}{2}$	6	$6\frac{1}{8}$	2	8	$7\frac{9}{16}$
$1\frac{1}{4}$	$2\frac{1}{2}$	$4\frac{1}{4}$	2	$3\frac{1}{4}$	$5\frac{3}{16}$	$2\frac{1}{2}$	6	$7\frac{1}{8}$
$1\frac{1}{4}$	3	$4\frac{5}{8}$	2	$3\frac{1}{2}$	$5\frac{3}{16}$	3	6	$7\frac{7}{16}$
$1\frac{1}{4}$	4	$5\frac{1}{8}$	2	$3\frac{3}{4}$	$5\frac{9}{16}$	4	8	$9\frac{15}{16}$
$1\frac{1}{4}$	$4\frac{1}{2}$	$5\frac{3}{16}$						

TABLE 26

General Dimensions of "York" Ells, Crosses and Tees

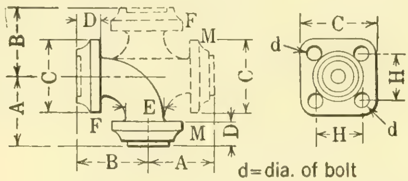
Oval Flange



All Dimensions Given in Inches

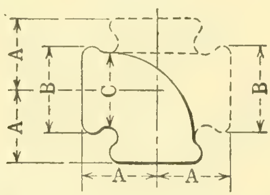
Size	*A*	*B*	*C*	*D*	*E*	*G*	*H*	*d*
$\frac{1}{4}$	$2\frac{7}{16}$	$2\frac{3}{8}$	$1\frac{3}{4}$	$\frac{7}{8}$	1	$3\frac{1}{2}$	$1\frac{1}{8}$	$\frac{1}{2}$
$\frac{3}{8}$ and $\frac{1}{2}$	$2\frac{13}{16}$	$2\frac{3}{4}$	$2\frac{1}{8}$	1	$1\frac{1}{4}$	$3\frac{3}{4}$	$1\frac{5}{16}$	$\frac{1}{2}$
$\frac{3}{4}$	$3\frac{5}{16}$	$3\frac{1}{4}$	$2\frac{3}{16}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$4\frac{1}{8}$	$1\frac{3}{8}$	$\frac{5}{8}$
1	$3\frac{9}{16}$	$3\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$4\frac{1}{2}$	$1\frac{9}{16}$	$\frac{5}{8}$
$1\frac{1}{4}$	$3\frac{13}{16}$	$3\frac{3}{4}$	$2\frac{5}{8}$	$1\frac{3}{8}$	$1\frac{7}{8}$	$4\frac{7}{8}$	$1\frac{21}{32}$	$\frac{5}{8}$

Square Flange



Size	*A*	*B*	*C*	*D*	*E*	*H*	*d*
$\frac{1}{2}$	$2\frac{13}{16}$	$2\frac{3}{4}$	3	1	$1\frac{1}{4}$	$1\frac{7}{8}$	$\frac{1}{2}$
1	$3\frac{9}{16}$	$3\frac{1}{2}$	$3\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{5}{8}$	$2\frac{1}{4}$	$\frac{1}{2}$
$1\frac{1}{4}$	$3\frac{13}{16}$	$3\frac{3}{4}$	$3\frac{3}{4}$	$1\frac{5}{16}$	$1\frac{13}{16}$	$2\frac{13}{32}$	$\frac{5}{8}$
$1\frac{1}{2}$	$4\frac{5}{16}$	$4\frac{1}{4}$	4	$1\frac{7}{16}$	$2\frac{3}{16}$	$2\frac{5}{8}$	$\frac{5}{8}$
2	$4\frac{5}{16}$	$4\frac{1}{4}$	$4\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{3}{4}$	$3\frac{1}{16}$	$\frac{5}{8}$
$2\frac{1}{2}$	$6\frac{1}{16}$	6	$5\frac{5}{8}$	$1\frac{7}{8}$	$3\frac{5}{16}$	4	$\frac{3}{4}$
3	$6\frac{1}{16}$	6	$6\frac{1}{8}$	$1\frac{7}{8}$	$3\frac{15}{16}$	$4\frac{1}{8}$	$\frac{3}{4}$
$3\frac{1}{2}$	$6\frac{9}{16}$	$6\frac{1}{2}$	$6\frac{5}{8}$	$1\frac{7}{8}$	$4\frac{1}{2}$	$4\frac{9}{16}$	$\frac{3}{4}$
4	$7\frac{1}{16}$	7	$7\frac{1}{8}$	$1\frac{7}{8}$	5	5	$\frac{7}{8}$
5	$7\frac{9}{16}$	$7\frac{1}{2}$	8	2	$6\frac{1}{4}$	$5\frac{7}{8}$	$\frac{7}{8}$

Screw End

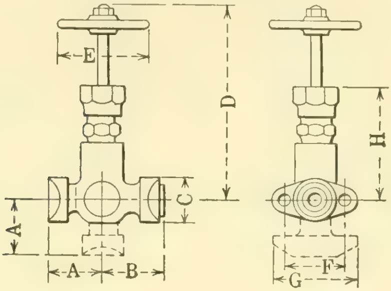


Size	*A*	*B*	*C*	Size	*A*	*B*	*C*	Size	*A*	*B*	*C*
$\frac{1}{4}$	1	$1\frac{1}{8}$	1	$1\frac{1}{4}$	$2\frac{1}{8}$	$2\frac{21}{32}$	$2\frac{5}{16}$	$3\frac{1}{2}$	4	$5\frac{3}{4}$	$5\frac{1}{4}$
$\frac{3}{8}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{2}$	$2\frac{1}{2}$	3	$2\frac{5}{8}$	4	$4\frac{3}{8}$	$6\frac{3}{8}$	$5\frac{5}{8}$
$\frac{1}{2}$	$1\frac{5}{16}$	$1\frac{15}{32}$	$1\frac{5}{16}$	2	$2\frac{7}{8}$	$3\frac{5}{8}$	$3\frac{1}{8}$	5	$5\frac{1}{4}$	$7\frac{3}{4}$	$6\frac{3}{4}$
$\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{13}{16}$	$1\frac{9}{16}$	$2\frac{1}{2}$	$3\frac{1}{4}$	$4\frac{1}{4}$	$3\frac{5}{8}$	6	$6\frac{1}{4}$	9	8
1	$1\frac{13}{16}$	$2\frac{3}{16}$	$1\frac{15}{16}$	3	$3\frac{5}{8}$	5	$4\frac{1}{2}$				

TABLE 27

Principal Dimensions of "York" Ammonia Valves

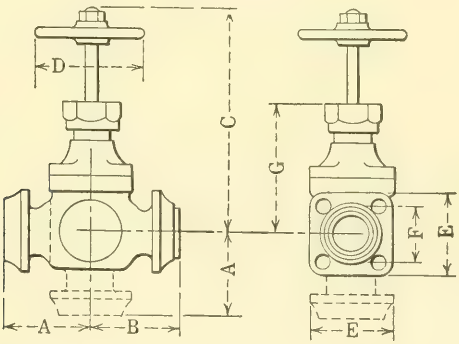
Oval Flange



All Dimensions Given in Inches

Size	A	B	C	D, Closed	D, Open	E	F	G	H
$\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{7}{16}$	$1\frac{3}{4}$	$7\frac{7}{8}$	$8\frac{1}{4}$	4	$2\frac{1}{4}$	$3\frac{1}{2}$	$4\frac{5}{8}$
$\frac{3}{8}$	$2\frac{3}{4}$	$2\frac{13}{16}$	$2\frac{1}{8}$	$8\frac{5}{8}$	$9\frac{1}{8}$	4	$2\frac{5}{8}$	$3\frac{3}{4}$	5
$\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{13}{16}$	$2\frac{1}{8}$	$8\frac{5}{8}$	$9\frac{1}{8}$	4	$2\frac{5}{8}$	$3\frac{3}{4}$	5
$\frac{3}{4}$	$3\frac{1}{4}$	$3\frac{5}{16}$	$2\frac{3}{16}$	9	$9\frac{11}{16}$	4	$2\frac{3}{4}$	$4\frac{1}{8}$	$5\frac{11}{16}$
1	$3\frac{1}{2}$	$3\frac{9}{16}$	$2\frac{1}{2}$	$9\frac{1}{2}$	$10\frac{1}{4}$	4	$3\frac{1}{8}$	$4\frac{1}{2}$	$5\frac{7}{8}$
$1\frac{1}{4}$	$3\frac{3}{4}$	$3\frac{13}{16}$	$2\frac{5}{8}$	$10\frac{13}{16}$	$11\frac{1}{4}$	$5\frac{1}{2}$	$3\frac{5}{16}$	$4\frac{7}{8}$	$6\frac{15}{16}$

Square Flange

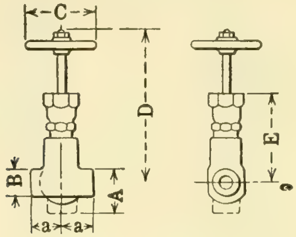


Size	A	B	C, Closed	C, Open	D	E	F	G
$1\frac{1}{4}$	$3\frac{3}{4}$	$3\frac{13}{16}$	$8\frac{9}{16}$	$9\frac{1}{16}$	$4\frac{3}{4}$	$3\frac{3}{4}$	$2\frac{3}{8}$	$6\frac{7}{16}$
$1\frac{1}{2}$	$4\frac{1}{4}$	$4\frac{5}{16}$	$10\frac{13}{16}$	$11\frac{1}{4}$	$5\frac{1}{2}$	4	$2\frac{5}{8}$	$6\frac{15}{16}$
2	$4\frac{1}{4}$	$4\frac{5}{16}$	$12\frac{1}{2}$	$13\frac{1}{16}$	$6\frac{3}{4}$	$4\frac{1}{2}$	$3\frac{1}{16}$	$7\frac{3}{16}$
$2\frac{1}{2}$	6	$6\frac{1}{16}$	$13\frac{5}{32}$	$13\frac{13}{16}$	$6\frac{3}{4}$	$5\frac{5}{8}$	4	$7\frac{13}{16}$
3	6	$6\frac{1}{16}$	$14\frac{5}{16}$	$15\frac{5}{16}$	8	$6\frac{1}{8}$	$4\frac{1}{8}$	$8\frac{15}{16}$
$3\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{9}{16}$	$14\frac{7}{8}$	$15\frac{7}{8}$	8	$6\frac{5}{8}$	$4\frac{9}{16}$	$9\frac{7}{16}$
4	7	$7\frac{1}{16}$	$16\frac{1}{4}$	$17\frac{3}{8}$	10	$7\frac{1}{8}$	5	$9\frac{3}{4}$
5	$7\frac{1}{2}$	$7\frac{9}{16}$	18	$19\frac{3}{8}$	12	8	$5\frac{7}{8}$	$11\frac{1}{4}$

TABLE 28

Principal Dimensions of "York" Ammonia Valves

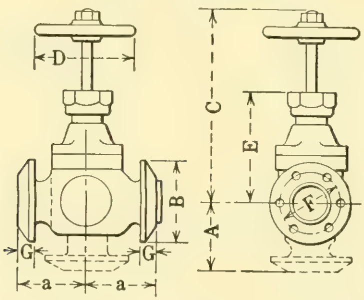
Screw Valves



All Dimensions Given in Inches

Size	*A*	*B*	*C*	*D*, Closed	*D*, Open	*E*
1/4	1 5/8	1 3/8	4	7 7/8	8 1/4	4 5/8
3/8	2	1 13/16	4	8 5/8	9 1/8	5
1/2	2	1 13/16	4	8 5/8	9 1/8	5
3/4	2 1/8	2 1/16	4	9	9 11/16	5 11/16
1	2 1/4	2 1/4	4	9 1/2	10 1/4	5 7/8
1 1/4	2 7/8	2 3/4	5 1/2	10 13/16	11 1/4	6 15/16
1 1/2	3	3	5 1/2	10 13/16	11 1/4	6 15/16
2	3 1/2	3 5/8	6 3/4	12 1/2	13 1/16	7 3/16
2 1/2	4 1/4	4 3/8	6 3/4	13 5/32	13 3/16	7 13/16
3	4 3/4	5	8	14 5/16	15 5/16	8 15/16

Round Flange Valves

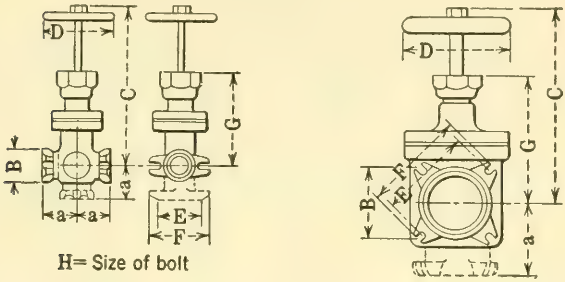


Size	*A*	*B*	*C*, Closed	*C*, Open	*D*	*E*	*F*	*G*	Bolts
2 1/2	6	7 1/8	15 1/8	15 7/8	6 1/4	10	5 1/2	1 3/8	6–3/4
3	6	7 7/8	14 5/16	15 5/16	8	8 15/16	6 1/8	1 1/2	6–3/4
3 1/2	6 1/2	8 5/8	14 7/8	15 7/8	8	9 7/16	6 7/8	1 1/2	8–3/4
4	7	9 3/8	16 1/4	17 3/8	10	9 3/4	7 5/8	1 5/8	8–3/4
4 1/2	7 1/2	9 3/4	....	....	....	....	8	1 11/16	8–3/4
5	7 1/2	10 1/4	18	19 3/8	12	11 1/4	8 1/2	1 3/4	8–3/4
6	8 1/2	11 1/2	20 1/16	21 11/16	14	13 5/16	9 3/4	1 7/8	8–3/4
7	10	12 1/2	21	22 3/4	14	14 1/4	10 3/4	1 15/16	8–7/8
8	12	14 1/4	22 3/8	24 1/2	14	16 1/16	12	2	8–7/8
10	14	16 1/2	24 13/16	27 3/16	18	18	14 1/4	2	12–7/8
12	16	18 3/4	27 11/16	30 15/16	18	20 9/16	16 1/2	2 1/8	12–1

TABLE 29

General Dimensions of "York" Gland End Fittings

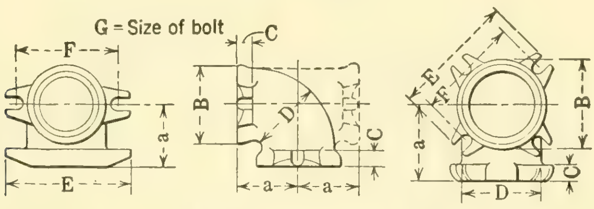
Globe and Angle Valves



Dimensions below Heavy Line Indicate Four Bolt Fittings
All Dimensions Given in Inches

Size	*A*	*B*	*C*, Closed	*C*, Open	*D*	*E*	*F*	*G*	*H*
1/2	2 1/4	1 3/4	8 5/8	9 1/8	4	2 1/2	3 1/2	5	1/2
3/4	2 3/8	2 1/8	9	9 11/16	4	2 7/8	3 7/8	5 11/16	1/2
1	2 1/2	2 3/8	9 1/2	10 1/4	4	3 1/8	4 1/8	5 7/8	1/2
1 1/4	3 1/8	2 7/8	10 13/16	11 1/4	5 1/2	3 1/2	4 1/2	6 15/16	1/2
1 1/2	3 1/4	3 1/8	10 13/16	11 1/4	5 1/2	3 5/8	4 5/8	6 15/16	1/2
2	3 3/4	3 5/8	12 1/2	13 1/16	6 3/4	4 1/4	5 1/4	7 3/16	1/2
2 1/2	4 1/2	4 1/4	13 5/32	13 13/16	6 3/4	5	6 1/4	7 13/16	5/8
3	5	5	14 5/16	15 5/16	8	6 1/4	7 3/4	8 15/16	3/4
3 1/2	5 1/2	5 1/2	14 7/8	15 7/8	8	6 1/2	7 5/8	9 7/16	5/8
4	6	6 1/4	16 1/2	17 3/8	10	7 1/4	8 3/4	9 3/4	3/4
5	7	7 1/2	18	19 3/8	12	8 1/2	10	11 1/4	3/4

Tees and Ells

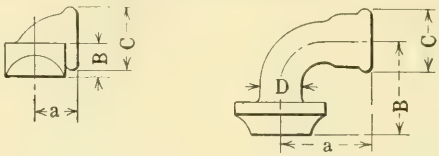


Size	*A*	*B*	*C*	*D*	*E*	*F*	*G*
1/2	1 5/8	1 3/4	5/8	1 1/2	3 1/2	2 1/2	1/2
3/4	1 3/4	2 1/8	3/4	1 13/16	3 7/8	2 7/8	1/2
1	1 7/8	2 3/8	3/4	2 1/16	4 1/8	3 1/8	1/2
1 1/4	2 1/8	2 7/8	3/4	2 1/2	4 1/2	3 1/2	1/2
1 1/2	2 3/8	3 1/8	3/4	2 3/4	4 5/8	3 5/8	1/2
2	2 7/8	3 5/8	7/8	3 3/16	5 1/4	4 1/4	1/2
2 1/2	3 1/2	4 1/4	1 1/8	3 3/4	6 1/4	5	5/8
3	3 3/4	5	1 1/8	4 5/8	7 3/4	6 1/4	3/4
3 1/2	4 3/4	5 1/2	1 1/8	5 1/16	7 5/8	6 1/2	5/8
4	5 1/4	6 1/4	1 1/4	5 5/8	8 3/4	7 1/4	3/4
5	6	7 1/2	1 1/2	6 7/8	10	8 1/2	3/4

TABLE 30

General Dimensions of "York" Screw and Flange Ells and Tees

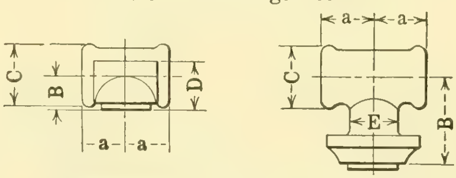
Screw and Flange Ells



All Flange Dimensions Standard Except as Noted
Dimensions above Heavy Line Are for Oval, below for Square

Size	*A*	*B*	*C*	*D*
1/4	1	3/4	1 1/8	
3/8 and 1/2	1 1/8	7/8	1 15/16	
3/4	1 1/4	1	1 13/16	
1	1 1/2	1 1/4	2 3/16	
1 1/4	1 3/4	1 3/8	2 21/32	
1/2	2	2 3/8	1 9/16	1 1/8
1	2 3/4	3	2 3/16	1 1/2
1 1/4	3 3/4	3 3/4	2 21/32	1 13/16
1 1/2	4 1/4	4 1/4	3	2 3/16
2	4 1/4	4 1/4	3 1/2	2 3/4
2 1/2	6	6	4 1/8	3 5/16
3	6	6	4 7/8	3 15/16
3 1/2	6 1/2	6 1/2	5 5/8	4 1/2
4	7	7	6 3/8	5
5	7 1/2	7 1/2	7 3/4	6 1/4

Screw and Flange Tees

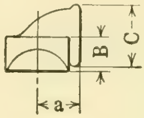


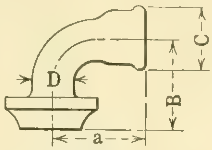
Size	*A*	*B*	*C*	*D*	*E*
1/4	1	3/4	1 1/8	7/8	
3/8 and 1/2	1 1/8	15/16	1 15/32	1 1/16	
3/4	1 1/4	1 1/16	1 13/16	1 7/16	
1	1 1/2	1 5/16	2 3/16	1 13/16	
1 1/4	1 3/4	1 7/16	2 21/32	2 1/16	
1/2					
1					
1 1/4	2 1/8	3 13/16	2 21/32	....	1 13/16
1 1/2	2 1/2	4 5/16	3	....	2 3/16
2	2 7/8	4 5/16	3 5/8	....	2 3/4
2 1/2	3 1/4	6 1/16	4 1/4	....	3 5/16
3	3 5/8	6 1/16	5	....	3 15/16
3 1/2	4	6 9/16	5 3/4	....	4 1/2
4	4 3/8	7 1/16	6 3/8	....	5
5	5 1/4	7 9/16	7 3/4	....	6 1/4

TABLE 31

General Dimensions of "York" Screw and Flange Ells and Tees

Screw and Flange Ells

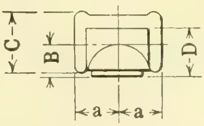


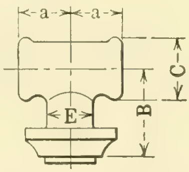


All Flange Dimensions Standard Except as Noted
Dimensions above Heavy Line Are for Oval, below for Square

Size	*A*	*B*	*C*	*D*
$\frac{1}{4}$	1	$\frac{3}{4}$	$1\frac{1}{8}$	
$\frac{3}{8}$ and $\frac{1}{2}$	$1\frac{1}{8}$	$\frac{7}{8}$	$1\frac{15}{32}$	
$\frac{3}{4}$	$1\frac{1}{4}$	1	$1\frac{13}{16}$	
1	$1\frac{1}{2}$	$1\frac{1}{4}$	$2\frac{3}{16}$	
$1\frac{1}{4}$	$1\frac{3}{4}$	$1\frac{3}{8}$	$2\frac{21}{32}$	
$\frac{1}{2}$	2	$2\frac{3}{8}$	$1\frac{9}{16}$	$1\frac{1}{8}$
1	$2\frac{3}{4}$	3	$2\frac{3}{16}$	$1\frac{1}{2}$
$1\frac{1}{4}$	$3\frac{3}{4}$	$3\frac{3}{4}$	$2\frac{21}{32}$	$1\frac{13}{16}$
$1\frac{1}{2}$	$4\frac{1}{4}$	$4\frac{1}{4}$	3	$2\frac{3}{16}$
2	$4\frac{1}{4}$	$4\frac{1}{4}$	$3\frac{1}{2}$	$2\frac{3}{4}$
$2\frac{1}{2}$	6	6	$4\frac{1}{8}$	$3\frac{5}{16}$
3	6	6	$4\frac{7}{8}$	$3\frac{15}{16}$
$3\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$5\frac{5}{8}$	$4\frac{1}{2}$
4	7	7	$6\frac{3}{8}$	5
5	$7\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{3}{4}$	$6\frac{1}{4}$

Screw and Flange Tees

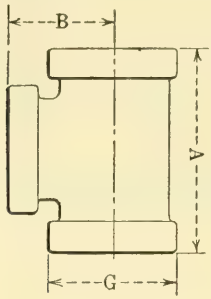




Size	*A*	*B*	*C*	*D*	*E*
$\frac{1}{4}$	1	$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{7}{8}$	
$\frac{3}{8}$ and $\frac{1}{2}$	$1\frac{1}{8}$	$\frac{15}{16}$	$1\frac{15}{32}$	$1\frac{1}{16}$	
$\frac{3}{4}$	$1\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{13}{16}$	$1\frac{7}{16}$	
1	$1\frac{1}{2}$	$1\frac{5}{16}$	$2\frac{3}{16}$	$1\frac{13}{16}$	
$1\frac{1}{4}$	$1\frac{3}{4}$	$1\frac{7}{16}$	$2\frac{21}{32}$	$2\frac{1}{16}$	
$\frac{1}{2}$					
1					
$1\frac{1}{4}$	$2\frac{1}{8}$	$3\frac{13}{16}$	$2\frac{21}{32}$	....	$1\frac{13}{16}$
$1\frac{1}{2}$	$2\frac{1}{2}$	$4\frac{5}{16}$	3	....	$2\frac{3}{16}$
2	$2\frac{7}{8}$	$4\frac{5}{16}$	$3\frac{5}{8}$	....	$2\frac{3}{4}$
$2\frac{1}{2}$	$3\frac{1}{4}$	$6\frac{1}{16}$	$4\frac{1}{4}$	....	$3\frac{5}{16}$
3	$3\frac{5}{8}$	$6\frac{1}{16}$	5	....	$3\frac{15}{16}$
$3\frac{1}{2}$	4	$6\frac{9}{16}$	$5\frac{3}{4}$	....	$4\frac{1}{2}$
4	$4\frac{3}{8}$	$7\frac{1}{16}$	$6\frac{3}{8}$	....	5
5	$5\frac{1}{4}$	$7\frac{9}{16}$	$7\frac{3}{4}$	....	$6\frac{1}{4}$

TABLE 32

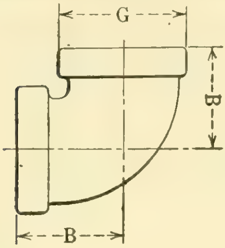
Frick Ammonia Tees—C. I. Screw



Size, Inches	A, Inches	B, Inches	G, Inches	Size, Inches	A, Inches	B, Inches	G, Inches
$\frac{1}{4}$ to $\frac{1}{4}\times\frac{1}{4}$	$1\frac{3}{4}$	$\frac{7}{8}$	$1\frac{1}{8}$	$\frac{1}{4}$ to $\frac{1}{4}\times1\frac{1}{4}$	$4\frac{1}{4}$	$2\frac{1}{8}$	$2\frac{5}{8}$
$\frac{3}{8}$ to $\frac{3}{8}\times\frac{3}{8}$	2	1	$1\frac{1}{4}$	$1\frac{1}{2}$ to $1\frac{1}{2}\times1\frac{1}{2}$	5	$2\frac{1}{2}$	3
$\frac{3}{8}$ to $\frac{1}{4}\times\frac{1}{4}$	2	1	$1\frac{1}{4}$	$1\frac{1}{2}$ to $\frac{1}{4}\times\frac{1}{4}$	5	$2\frac{1}{2}$	3
$\frac{1}{4}$ to $\frac{1}{4}\times\frac{3}{8}$	2	1	$1\frac{1}{4}$	$\frac{1}{4}$ to $\frac{1}{4}\times1\frac{1}{2}$	5	$2\frac{1}{2}$	3
$\frac{1}{2}$ to $\frac{1}{2}\times\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{2}$	2 to 2 $\times2$	$5\frac{3}{4}$	$2\frac{7}{8}$	$3\frac{1}{2}$
$\frac{1}{2}$ to $\frac{1}{4}\times\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{2}$	2 to $\frac{1}{4}\times\frac{1}{4}$	$5\frac{3}{4}$	$2\frac{7}{8}$	$3\frac{1}{2}$
$\frac{1}{4}$ to $\frac{1}{4}\times\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$\frac{1}{4}$ to $\frac{1}{4}\times2$	$5\frac{3}{4}$	$2\frac{7}{8}$	$3\frac{1}{2}$
$\frac{3}{4}$ to $\frac{3}{4}\times\frac{3}{4}$	3	$1\frac{1}{2}$	$1\frac{13}{16}$	$2\frac{1}{2}$ to $2\frac{1}{2}\times2\frac{1}{2}$	$10\frac{1}{2}$	$5\frac{1}{4}$	$4\frac{7}{8}$
$\frac{3}{4}$ to $\frac{1}{4}\times\frac{1}{4}$	3	$1\frac{1}{2}$	$1\frac{13}{16}$	$2\frac{1}{2}$ to 2 $\times2$	$10\frac{1}{2}$	$5\frac{1}{4}$	$4\frac{7}{8}$
$\frac{1}{4}$ to $\frac{1}{4}\times\frac{3}{4}$	3	$1\frac{1}{2}$	$1\frac{13}{16}$	3 to 3 $\times3$	$10\frac{1}{2}$	$5\frac{1}{4}$	$5\frac{1}{2}$
1 to 1 $\times1$	$3\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$3\frac{1}{2}$ to $3\frac{1}{2}\times3\frac{1}{2}$	$11\frac{1}{2}$	$5\frac{3}{4}$	6
1 to $\frac{1}{4}\times\frac{1}{4}$	$3\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{8}$	4 to 4 $\times4$	$12\frac{1}{2}$	$6\frac{1}{4}$	$6\frac{3}{4}$
$\frac{1}{4}$ to $\frac{1}{4}\times1$	$3\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$4\frac{1}{2}$ to $4\frac{1}{2}\times4\frac{1}{2}$	$13\frac{1}{2}$	$6\frac{3}{4}$	$7\frac{3}{8}$
$1\frac{1}{4}$ to $1\frac{1}{4}\times1\frac{1}{4}$	$4\frac{1}{4}$	$2\frac{1}{8}$	$2\frac{5}{8}$	5 to 5 $\times5$	$14\frac{1}{2}$	$7\frac{1}{4}$	8
$1\frac{1}{4}$ to $\frac{1}{4}\times\frac{1}{4}$	$4\frac{1}{4}$	$2\frac{1}{8}$	$2\frac{5}{8}$				

TABLE 32—*Continued*

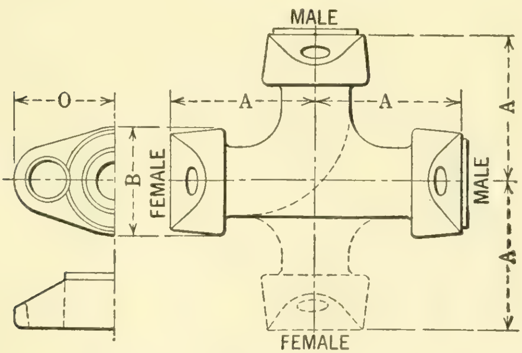
AMMONIA ELLS—C. I. SCREW



Size, Inches	B, Inches	G, Inches	Size, Inches	B, Inches	G, Inches	Size, Inches	B, Inches	G, Inches
$\frac{1}{4}$	$\frac{7}{8}$	$1\frac{1}{8}$	1	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\times\frac{1}{4}$	$2\frac{7}{8}$	$3\frac{1}{2}$
$\frac{3}{8}$	1	$1\frac{1}{4}$	$1\times\frac{1}{4}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{1}{2}$	$5\frac{1}{4}$	$4\frac{7}{8}$
$\frac{3}{8}\times\frac{1}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{4}$	$2\frac{1}{8}$	$2\frac{5}{8}$	3	$5\frac{1}{4}$	$5\frac{1}{2}$
$\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}\times\frac{1}{4}$	$2\frac{1}{8}$	$2\frac{5}{8}$	$3\frac{1}{2}$	$5\frac{3}{4}$	6
$\frac{1}{2}\times\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{1}{2}$	3	4	$6\frac{1}{4}$	$6\frac{3}{4}$
$\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{13}{16}$	$1\frac{1}{2}\times\frac{1}{4}$	$2\frac{1}{2}$	3	$4\frac{1}{2}$		
$\frac{3}{4}\times\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{13}{16}$	2	$2\frac{7}{8}$	$3\frac{1}{2}$	5	$7\frac{1}{4}$	8

TABLE 33

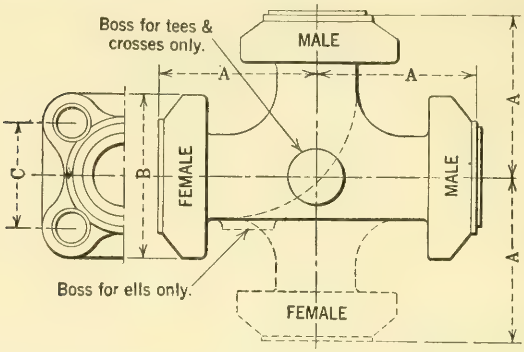
FRICK AMMONIA ELLS, TEES AND CROSSES—OVAL FLANGE



Size, Inches	A, Inches	B, Inches	O, Inches	Size, Inches	A, Inches	B, Inches	O, Inches
$\frac{1}{4}$	$2\frac{1}{8}$	$1\frac{3}{4}$	$1\frac{3}{4}$	$\frac{3}{4}$	3	$2\frac{3}{16}$	$2\frac{1}{16}$
$\frac{3}{8}$	$2\frac{3}{8}$	$2\frac{1}{16}$	$1\frac{7}{8}$	1	$3\frac{1}{4}$	$2\frac{7}{16}$	$2\frac{1}{4}$
$\frac{1}{2}$	$2\frac{3}{8}$	$2\frac{1}{16}$	$1\frac{7}{8}$				

TABLE 34

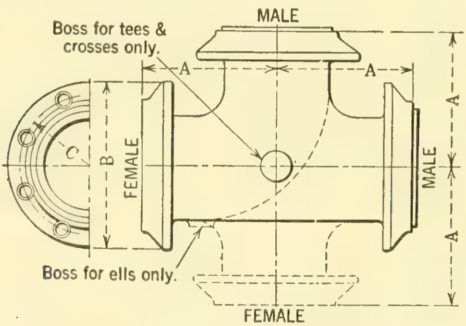
Frick Ammonia Ells, Tees and Crosses—Square Flange



Size, Inches	A, Inches	B, Inches	C, Inches	Bolts, Inches	Size, Inches	A, Inches	B, Inches	C, Inches	Bolts, Inches
$1\frac{1}{4}$	$3\frac{3}{4}$	$3\frac{3}{4}$	$2\frac{3}{8}$	$4-\frac{5}{8}\times2\frac{7}{8}$	3	6	$6\frac{1}{8}$	$4\frac{1}{8}$	$4-\frac{3}{4}\times4\frac{1}{8}$
$1\frac{1}{2}$	$4\frac{1}{4}$	4	$2\frac{5}{8}$	$4-\frac{5}{8}\times3$	$3\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{5}{8}$	$4\frac{9}{16}$	$4-\frac{3}{4}\times4\frac{3}{8}$
2	$4\frac{1}{4}$	$4\frac{1}{2}$	$3\frac{1}{16}$	$4-\frac{5}{8}\times3\frac{1}{4}$	4	7	$7\frac{1}{8}$	5	$4-\frac{7}{8}\times4\frac{3}{4}$
$2\frac{1}{2}$	6	$5\frac{5}{8}$	$3\frac{15}{16}$	$4-\frac{3}{4}\times3\frac{7}{8}$					

TABLE 35

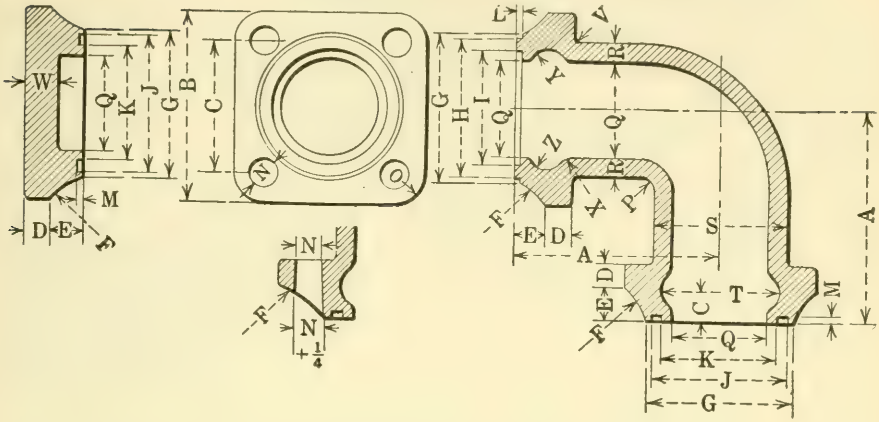
Frick Ammonia Ells, Tees and Crosses—Round Flange



Size, Inches	A, Inches	B, Inches	C, Inches	Bolts, Inches	Size, Inches	A, Inches	B, Inches	C, Inches	Bolts, Inches
5	$8\frac{1}{2}$	$10\frac{1}{4}$	$8\frac{1}{2}$	$8-\frac{5}{8}\times4\frac{3}{4}$	8	$12\frac{3}{4}$	$14\frac{1}{2}$	$12\frac{1}{2}$	$8-\frac{7}{8}\times6$
6	$9\frac{1}{2}$	$11\frac{1}{2}$	$9\frac{3}{4}$	$8-\frac{3}{4}\times5\frac{1}{8}$	10	16	17	15	$12-\frac{7}{8}\times5\frac{3}{4}$
7	$11\frac{1}{2}$	$13\frac{1}{2}$	$11\frac{1}{8}$	$8-\frac{7}{8}\times6$					

TABLE 36

Arctic Standard Square Flange Fittings



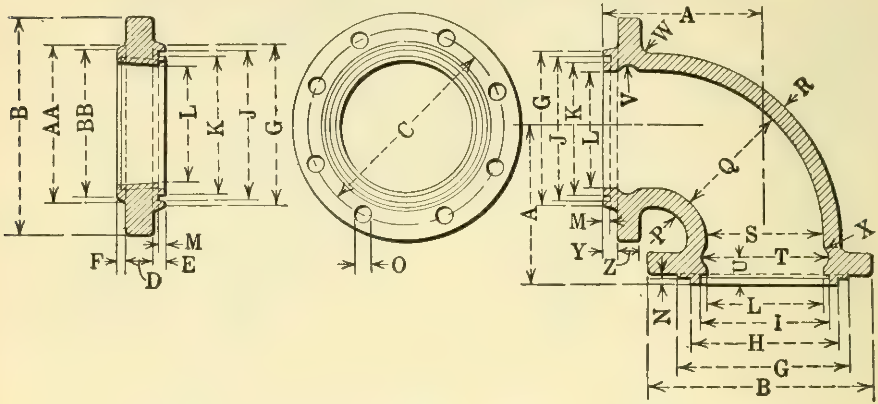
All Dimensions Given in Inches

Pipe Size	A	B	C	D	E	F	G	H	I	Diameter of Bolt	Length of Bolt	Flange to Flange
1/2	3	2 7/8	1 7/8	1/2	1/2	3	2 1/4	1 7/8	1 1/4	7/16	2 1/2	7/16 ×2
1 1/4	3 3/8	3 1/2	2 3/8	1/2	1/2	3	3	2 19/32	1 7/8	1/2	2 3/4	1/2 ×2 1/2
1 1/2	4	4	2 3/4	9/16	9/16	3	3 1/4	2 3/4	2	1/2	2 3/4	
2	4 3/4	4 1/2	3 1/16	5/8	5/8	3	3 3/4	3 1/4	2 1/2	5/8	3 1/4	5/8 ×3
2 1/2	6	5 1/2	3 7/8	3/4	7/8	3	4 1/8	3 25/32	3	3/4	4 1/4	
3	6 1/2	6	4 1/8	3/4	1	3	4 3/4	4 11/32	3 5/8	3/4	4 1/4	
3 1/2	7	6 1/2	4 1/2	3/4	1	3	5 1/2	4 7/8	4 3/32	3/4	4 1/4	
4	7 1/4	7	5	3/4	1	3	6	5 1/2	4 3/4	7/8	4 1/2	
4 1/2	7 1/2	7 1/2	5 1/2	3/4	1	3	6 5/8	6	5 1/4	7/8	4 1/2	
5	7 3/4	8	6	3/4	1	3	7 1/4	6 1/2	5 3/4	7/8	4 1/2	
	J	K	L	M	N	O	P	Q	R			
1/2	1 29/32	1 7/32	3/16	5/32	1/2	1/2	1/2	1	5/16	7/16	2 1/2	7/16 ×2
1 1/4	2 5/8	1 27/32	3/16	5/32	5/8	9/16	1/2	1 1/4	3/8	1/2	2 3/4	1/2 ×2 1/2
1 1/2	2 25/32	1 31/32	3/16	5/32	5/8	5/8	1/2	1 1/2	3/8	1/2	2 3/4	
2	3 9/32	2 15/32	3/16	5/32	3/4	23/32	1/2	2	7/16	5/8	3 1/4	5/8 ×3
2 1/2	3 13/16	2 31/32	3/16	5/32	7/8	13/16	1/2	2 1/2	1/2	3/4	4 1/4	
3	4 3/8	3 19/32	3/16	5/32	7/8	15/16	1/2	3	9/16	3/4	4 1/4	
3 1/2	4 29/32	4 1/16	3/16	5/32	7/8	1	1/2	3 1/2	9/16	3/4	4 1/4	
4	5 9/16	4 11/16	3/16	5/32	1	1	1/2	4	5/8	7/8	4 1/2	
4 1/2	6 1/16	5 3/16	3/16	5/32	1	1	1/2	4 1/2	5/8	7/8	4 1/2	
5	6 9/16	5 11/16	3/16	5/32	1	1	1/2	5	11/16	7/8	4 1/2	
	S	T	U	V	W	X	Y	Z				
1/2	1 5/8	1 5/8	5/8	1/4	.....	1/2	1/8	1/4	.....	7/16	2 1/2	7/16 ×2
1 1/4	2	1 5/8	3/4	1/4	.....	1/2	1/8	5/16	.....	1/2	2 3/4	1/2 ×2 1/2
1 1/2	2 1/4	2 1/8	3/4	1/4	.....	1/2	1/4	5/16	.....	1/2	2 3/4	
2	2 7/8	2 1/2	3/4	1/4	.....	7/8	1/4	3/8	.....	5/8	3 1/4	5/8 ×3
2 1/2	3 1/2	3 1/8	1	1/4	1	7/8	1/4	5/8	.....	3/4	4 1/4	
3	4 1/8	3 3/4	1	1/4	1	7/8	1/4	5/8	.....	3/4	4 1/4	
3 1/2	4 5/8	4 1/8	1	1/4	1 1/8	7/8	1/4	5/8	.....	3/4	4 1/4	
4	5 1/4	4 11/16	1	1/4	1 1/8	7/8	1/4	5/8	.....	7/8	4 1/2	
4 1/2	5 3/4	5 3/16	1	1/4	1 3/16	7/8	1/4	5/8	.....	7/8	4 1/2	
5	6 3/8	5 3/4	1	1/4	1 3/16	7/8	1/4	5/8	.....	7/8	4 1/2	

Center to face dimension, "A," applies to ells, tees, crosses, angle valves, and straight-way valves.
Condenser inlet header tee branches are 7 in. center to face.
Condenser outlet header tee branches are 6 in. center to face.
Center to face of reducing tees is governed by dimensions for largest opening.
Fittings and valves are male one end, female other end. Branches on all tees and crosses are male, except on condenser outlet header tees.

TABLE 37

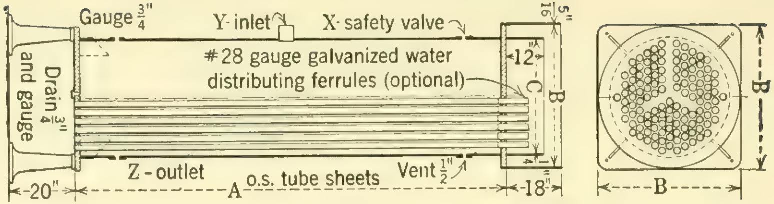
Arctic Standard Round Flanged Tongue and Groove Fittings



All Dimensions Except under Heading "Number of Bolts" Given in Inches

Size of Pipe	Material of Flange	A	B	C	D	E	F	G	H	I	J	Number of Bolts	Diameter of Bolt	Length of Bolt: Fl. to fl.	Length of Bolt: Fl. to fit.	Length of Bolt: Fit. to fit.
$4\frac{1}{2}$	ST.	$7\frac{1}{2}$	$9\frac{3}{4}$	8	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{16}$	$6\frac{5}{8}$	6	$5\frac{1}{4}$	$6\frac{1}{16}$	8	$\frac{3}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$	$3\frac{3}{4}$
5	ST.	8	$10\frac{1}{4}$	$8\frac{1}{2}$	$\frac{5}{8}$	$\frac{13}{16}$	$\frac{1}{2}$	$7\frac{1}{4}$	$6\frac{1}{2}$	$5\frac{3}{4}$	$6\frac{9}{16}$	8	$\frac{3}{4}$	$3\frac{3}{4}$	$3\frac{3}{4}$	4
6	CI.	$8\frac{1}{2}$	$11\frac{3}{8}$	$9\frac{5}{8}$	$1\frac{1}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	$8\frac{1}{2}$	$7\frac{3}{4}$	7	$7\frac{13}{16}$	8	$\frac{3}{4}$	4	4	4
7	CI.	9	$12\frac{7}{8}$	$10\frac{7}{8}$	$1\frac{5}{16}$	$\frac{7}{16}$	$\frac{3}{8}$	$9\frac{5}{8}$	$8\frac{3}{4}$	8	$8\frac{13}{16}$	8	$\frac{7}{8}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$
8	CI.	$9\frac{1}{2}$	$13\frac{5}{8}$	$11\frac{7}{8}$	$1\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	$10\frac{1}{4}$	$9\frac{3}{4}$	9	$9\frac{13}{16}$	12	$\frac{3}{4}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$
		K	L	M	N	O	P	Q	R	S	T					
$4\frac{1}{2}$	ST.	$5\frac{3}{16}$	$4\frac{5}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{7}{8}$	$1\frac{1}{2}$	$4\frac{1}{2}$	$\frac{5}{8}$	$5\frac{3}{4}$	$5\frac{1}{4}$	8	$\frac{3}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$	$3\frac{3}{4}$
5	ST.	$5\frac{11}{16}$	$5\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{7}{8}$	$1\frac{1}{2}$	5	$\frac{11}{16}$	$6\frac{3}{8}$	$5\frac{3}{4}$	8	$\frac{3}{4}$	$3\frac{3}{4}$	$3\frac{3}{4}$	4
6	CI.	$6\frac{15}{16}$	$6\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{7}{8}$	$1\frac{1}{2}$	6	$\frac{3}{4}$	$7\frac{1}{2}$	$6\frac{13}{16}$	8	$\frac{3}{4}$	4	4	4
7	CI.	$7\frac{15}{16}$	$7\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	1	$1\frac{1}{2}$	7	$\frac{13}{16}$	$8\frac{5}{8}$	8	8	$\frac{7}{8}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$
8	CI.	$8\frac{15}{16}$	$8\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{7}{8}$	2	8	$\frac{7}{8}$	$9\frac{3}{4}$	$8\frac{7}{8}$	12	$\frac{3}{4}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$
		U	V	W	X	Y	Z	AA	BB							
$4\frac{1}{2}$	ST.	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{3}{4}$	$\frac{5}{16}$	$1\frac{1}{8}$	6	$5\frac{7}{8}$	...	...	8	$\frac{3}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$	$3\frac{3}{4}$
5	ST.	$\frac{7}{8}$	$\frac{7}{16}$	$\frac{1}{4}$	$\frac{15}{16}$	$\frac{3}{8}$	$1\frac{3}{16}$	$6\frac{1}{2}$	$6\frac{3}{8}$	...	...	8	$\frac{3}{4}$	$3\frac{3}{4}$	$3\frac{3}{4}$	4
6	CI.	$\frac{7}{8}$	$\frac{7}{16}$	$\frac{1}{4}$	1	$\frac{3}{8}$	$1\frac{1}{4}$	$7\frac{5}{8}$	$7\frac{1}{2}$	...	...	8	$\frac{3}{4}$	4	4	4
7	CI.	$\frac{7}{8}$	$\frac{7}{16}$	$\frac{1}{4}$	$1\frac{1}{16}$	$\frac{7}{16}$	$1\frac{5}{16}$	$8\frac{3}{4}$	$8\frac{5}{8}$	...	...	8	$\frac{7}{8}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$
8	CI.	1	$\frac{1}{2}$	$\frac{1}{4}$	$1\frac{1}{16}$	$\frac{1}{2}$	$1\frac{3}{8}$	$9\frac{3}{4}$	$9\frac{5}{8}$	...	...	12	$\frac{3}{4}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$

TABLE 38
STRUTHERS–WELLS CO. CONDENSERS

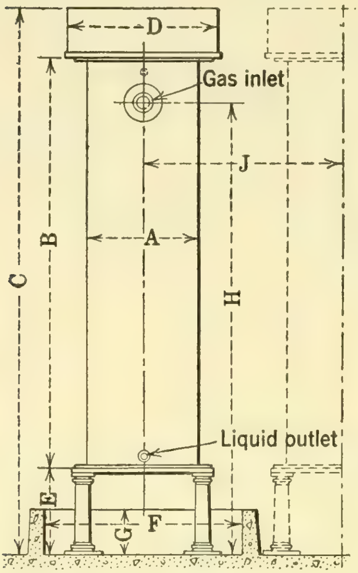


Outside Diameter, Inches	Thickness of Shell, Inches	Thickness of Tube Sheet, Inches	Number of Tubes	Length of Tubes	Dimensions A		Dimensions B	Dimensions C	S–V X	Inlet Y	Outlet Z	Tube Surface, Square Feet	Capacity of Refrigeration, Tons
				Ft.	Ft.	In.	Inches	Inches	Inches	Inches	Inches		
				6	5	$11\frac{1}{2}$						66	5.5
16	$\frac{1}{4}$	$1\frac{3}{8}$	22	8	7	$11\frac{1}{2}$	28	16	$\frac{3}{4}$	2	$\frac{3}{4}$	88	7.3
				10	9	$11\frac{1}{2}$						110	9.2
				8	7	$11\frac{1}{2}$						120	10.0
18	$\frac{1}{4}$	$1\frac{3}{8}$	30	10	9	$11\frac{1}{2}$	30	18	$\frac{3}{4}$	2	$\frac{3}{4}$	150	12.5
				12	11	$11\frac{1}{2}$						180	15.0
				10	9	$11\frac{1}{2}$						175	14.6
20	$\frac{5}{16}$	$1\frac{3}{8}$	35	12	11	$11\frac{1}{2}$	32	20	$\frac{3}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	210	17.5
				14	13	$11\frac{1}{2}$						245	20.4
				12	11	$11\frac{1}{2}$						265	22.1
22	$\frac{5}{16}$	$1\frac{3}{8}$	44	14	13	$11\frac{1}{2}$	34	22	$\frac{3}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	308	25.6
				16	15	$11\frac{1}{2}$						352	29.3
				14	13	$11\frac{1}{2}$						350	29.1
24	$\frac{3}{8}$	$1\frac{3}{8}$	50	16	15	$11\frac{1}{2}$	36	24	1	$2\frac{1}{2}$	1	400	33.3
				18	17	$11\frac{1}{2}$						450	37.5
				14	13	$11\frac{1}{2}$						435	36.3
26	$\frac{3}{8}$	$1\frac{3}{8}$	62	16	15	$11\frac{1}{2}$	38	26	1	$2\frac{1}{2}$	1	495	41.2
				18	17	$11\frac{1}{2}$						557	46.4

NOTE.—Tubes are 2 in. O. D. × No. 10 B. W. G. Charcoal Iron. Condenser electrically welded throughout unless otherwise specified. Test 500 lb. hydro. pressure.

TABLE 39

Carbondale Condensers



Refrigeration Capacity, Tons	A	B		C		D	E	F	
	Inches	Ft.	In.	Ft.	In.	Inches	Inches	Ft.	In.
24	26½	8	4¼	11	5	37½	23⅜	4	4
32	26½	11	1½	14	2¼	37½	23⅜	4	4
38	26½	13	1½	16	2¼	37½	23⅜	4	4
55	30	15	0¼	18	1	41	23⅜	4	6
62	30	17	0¼	20	1	41	23⅜	4	6
67	36¾	11	1½	14	4¼	47¾	25⅜	5	2
79	36¾	13	1½	16	4¼	47¾	25⅜	5	2
90	36¾	15	0¼	18	3	47¾	25⅓	5	2
106	46¼	11	1½	14	2¼	57¾	23⅜	5	6
125	46¼	13	1½	16	4¼	57¾	25⅜	5	6

Refrigeration Capacity, Tons	G	H		J		Gas	Liquor	Pump Out
	Inches	Ft.	In.	Ft.	In.	Inches	Inches	Inches
24	12	8	9⅝	3	8	1½	1	1
32	12	11	6⅞	3	8	2	1¼	1
38	12	13	6⅞	3	8	2	1¼	1
55	12	15	5⅝	4	0	2½	1¼	1
62	12	17	5⅝	4	0	2½	1¼	1
67	12	11	8⅞	4	6	2½	1½	1
79	12	13	8⅞	4	6	2½	1½	1
90	12	15	7⅝	4	6	3	2	1
106	12	11	6⅞	5	6	3	2	1
125	12	13	8⅞	5	6	3	2	1¼

TABLE 40

ATMOSPHERIC AMMONIA CONDENSERS 2-IN. PIPE—20 FT. LONG

Pipes High	Gallons per Minute per Coil	Initial Temperature of Condensing Water																				
		60 Deg. F.			65 Deg. F.			70 Deg. F.			75 Deg. F.			80 Deg. F.			85 Deg. F.			90 Deg. F.		
		Condenser pressure, pounds per square inch (gage)																				
		145	165	185	145	165	185	155	175	185	165	185	205	175	185	205	185	205	215	205	215	235
		Tons of refrigeration per 24 hours																				
8	20	6	7	8	5	6	7	5	6	7	5	6	7	5	6	7	4	5	6	4	5	6
	15	5	6	7	4	5	6	4	5	6	4	5	6	4	5	6	3	4	5	3	4	5
	10	4	5	6	3	4	5	3	4	5	3	4	5	3	4	5	2	3	4	2	3	4
12	20	7	8	9	6	7	8	6	7	8	6	7	8	6	7	8	6	7	8	6	7	8
	15	6	7	8	5	6	7	5	6	7	5	6	7	5	6	7	5	6	7	5	6	7
	10	5	6	7	4	5	6	4	5	6	4	5	6	4	5	6	4	5	6	4	5	6
16	25	8	10	12	7	8	10	7	8	10	7	8	9	7	8	9	7	8	9	7	8	9
	20	7	9	11	6	7	9	6	7	9	6	7	8	6	7	8	6	7	8	6	7	8
	15	6	8	10	5	6	8	5	6	8	5	6	7	5	6	7	5	6	7	5	6	7
20	25	10	12	14	9	10	12	9	10	12	9	10	12	9	10	11	8	9	10	8	9	10
	20	9	11	13	8	9	11	8	9	11	8	9	10	8	9	10	7	8	9	7	8	9
	15	8	10	12	7	8	10	7	8	10	7	8	9	7	8	9	6	7	8	6	7	8
24	30	12	13	15	11	12	14	11	12	13	11	12	13	10	11	12	10	11	12	10	11	12
	25	11	12	14	10	11	12	10	11	12	10	11	12	9	10	11	9	10	11	9	10	11
	20	10	11	12	9	10	11	9	10	11	9	10	11	8	9	10	8	9	10	8	9	10

For condensers 10 ft. long divide above total G.P.M. and tons of refrigeration by 2.

TABLE 41

CAPACITIES ATMOSPHERIC DRIP AMMONIA CONDENSERS 2-IN. PIPE—20 FT. LONG

Pipes per Stand.	Gallons per Minute per Coil	Initial Temperature of Condensing Water																				
		60 Deg. F.			65 Deg. F.			70 Deg. F.			75 Deg. F.			80 Deg. F.			85 Deg. F.			90 Deg. F.		
		Condenser pressure, pounds per square inch (gage)																				
		145	165	185	145	165	185	155	175	185	165	185	205	175	185	205	185	205	215	205	215	235
		Tons of refrigeration per 24 hours																				
12	10	5.6	8.0	10.1	4.1	6.4	8.4	3.6	5.8	6.8	3.2	5.2	7.2	2.6	3.6	5.6	2.0	4.0	4.9	2.4	3.2	5.1
	15	8.5	12.0	15.1	6.1	9.5	12.7	5.4	8.7	10.3	4.8	7.9	10.8	3.9	5.4	8.4	3.0	6.0	7.4	3.6	4.8	7.6
	20	11.3	16.0	20.2	8.2	12.7	16.9	7.3	11.7	13.7	6.4	10.5	14.4	5.2	7.3	11.2	4.1	8.0	9.8	4.7	6.4	10.2
	25	14.1	20.0	24.8	10.2	15.9	21.2	9.1	14.6	17.2	8.0	13.1	18.0	6.5	9.1	14.0	5.1	10.0	12.3	6.0	8.0	12.7
	30	16.5	20.0	....	12.2	17.9	....	10.9	16.8	18.8	9.6	15.8	19.4	7.8	10.9	16.5	6.1	12.0	14.8	7.1	9.6	15.3
	35	....	....	....	13.6	....	....	12.7	....	....	11.1	....	....	9.1	12.7	....	7.1	13.4	15.2	8.3	12.1	
	40	....	....	....	....	....	....	....	....	....	11.9	....	....	10.5	....	....	8.1	....	....	9.1		
16	25	14.1	20.0	25.0	10.2	15.9	21.2	9.1	14.6	17.2	8.0	13.1	18.0	6.5	9.1	14.0	5.1	10.0	12.3	6.0	8.0	12.7
	30	16.8	23.9	30.0	12.2	18.9	25.3	10.9	17.3	20.5	9.6	15.8	21.5	7.8	10.9	16.5	6.1	12.0	14.8	7.1	9.6	15.3
	35	19.6	27.4	33.4	14.2	22.0	29.0	12.6	20.2	23.9	11.0	18.3	25.0	9.1	12.7	19.5	7.1	14.0	17.2	8.2	11.2	17.6
	40	22.2	....	....	16.2	23.8	....	14.4	22.4	25.0	12.5	21.1	25.8	10.3	14.8	21.9	8.1	16.0	19.5	9.4	12.8	20.1
	45	....	....	....	18.2	....	....	16.3	....	....	14.2	....	....	11.6	16.4	....	9.1	18.0	20.2	10.6	14.4	20.6
	50	....	....	....	....	....	....	17.0	....	....	15.5	....	....	12.8	17.1	....	10.1	....	....	11.8	16.0	

For 10-ft. condensers divide the above capacities and G.P.M. by 2.

TABLE 42

CAPACITIES DOUBLE-PIPE AMMONIA CONDENSERS 1¼ and 2-IN. PIPE—20 FT. LONG

Pipe High	Gallons per Minute per Coil	Feet Head	Initial Temperature of Condensing Water																				
			60 Deg. F.			65 Deg. F.			70 Deg. F.			75 Deg. F.			80 Deg. F.			85 Deg. F.			90 Deg. F.		
			Condenser pressure, pounds per square inch (gage)																				
			145	165	185	165	185	195	175	185	205	185	205	215	185	205	215	185	205	215	205	215	235
			Tons of refrigeration per 24 hours																				
4	5	1	2.0	2.5	3.0	2.0	2.5	2.5	1.5	2.0	2.5	1.5	1.8	2.0	1.3	1.5	2.0	1.0	1.2	1.3	1.0	1.0	1.5
	10	3	2.5	4.0	5.0	3.0	3.5	4.0	2.5	3.0	3.5	2.5	2.8	3.0	2.0	2.5	3.0	1.5	2.0	2.3	1.5	1.7	2.0
	20	9	4.5	6.0	7.0	5.0	6.0	7.0	4.5	6.0	6.0	4.0	5.5	6.0	3.0	4.0	5.5	2.0	3.0	4.0	2.5	3.0	4.0
6	15	9	5.4	7.2	8.4	6.0	7.2	8.4	5.4	6.6	7.8	4.8	6.6	7.2	3.6	4.8	6.6	2.4	4.2	4.8	3.0	3.6	4.2
	25	22	8.2	10.5	13.0	9.0	11.3	13.0	8.2	9.8	12.0	7.5	9.7	10.5	6.0	8.3	9.7	3.8	6.0	6.8	4.5	5.3	6.7
	35	43	10.0	12.0	16.0	11.0	12.0	15.0	10.0	12.0	14.0	9.8	12.0	14.0	7.0	9.8	12.0	5.0	7.7	9.1	5.6	6.3	9.1
8	15	11	7.8	9.8	12.0	8.5	10.0	12.0	8.0	9.1	11.0	6.6	9.1	10.0	5.2	7.1	9.1	3.8	5.8	6.5	3.9	4.2	6.5
	25	30	12.0	15.0	19.0	12.0	16.0	18.0	12.0	14.0	16.0	10.0	14.0	15.0	7.8	11.0	14.0	5.2	8.5	10.0	5.8	7.1	9.1
	35	58	14.0	18.0	22.0	14.0	19.0	20.0	14.0	16.0	19.0	12.0	16.0	18.0	9.8	13.0	16.0	6.5	10.0	15.0	7.0	8.5	12.0
10	15	14	9.0	11.0	13.0	9.0	11.0	13.0	9.0	10.0	12.0	8.0	10.0	11.0	6.0	8.4	10.0	4.0	6.6	7.3	4.5	6.7	8.0
	25	38	13.0	18.0	22.0	13.0	18.0	20.0	14.0	15.0	19.0	12.0	16.0	18.0	8.8	13.0	16.0	6.2	9.6	11.0	6.5	9.5	12.0
	35	73	15.0	20.0	24.0	16.0	21.0	23.0	15.0	18.0	22.0	14.0	18.0	20.0	11.0	14.0	18.0	7.3	11.0	13.0	8.0	12.0	16.0
12	15	17	9.0	12.0	15.0	9.7	13.0	15.0	9.0	11.0	13.0	9.0	11.0	12.0	6.5	9.0	12.0	5.0	7.1	8.5	4.0	6.0	8.0
	25	45	14.0	19.0	23.0	16.0	20.0	22.0	14.0	16.0	20.0	13.0	17.0	19.0	9.3	14.0	17.0	6.6	10.0	12.0	7.0	9.0	12.0
	35	88	17.0	23.0	29.0	19.0	25.0	27.0	16.0	20.0	25.0	16.0	22.0	23.0	12.0	17.0	21.0	9.0	13.0	14.0	8.0	12.0	16.0

CHAPTER V

THE AUTOMATIC REFRIGERATING MACHINE

The larger refrigerating machine would necessarily require an operator for part or full time in order to maintain proper operating conditions and to make such adjustments as the machinery requires, but the smaller machines have to be designed in such a way that they will be automatic. This means that they must be capable of starting and stopping under the conditions of temperature in the refrigerated tank or room and that there be a regulator for the water (if water cooling is used) and for the suction pressure, in addition to various other safeguards. Under these conditions the only need for an operating engineer is to prevent clogging of valves, to maintain satisfactory lubrication and to watch against leaks or such troubles as might interfere with the proper functioning of the refrigerant. Development of such automatic machines results in an immediate expansion in distribution, not only to the small retail trader of meats and provisions, the florist and caterer, but to the soda fountain and small ice cream retailer and to household trade. Automatic refrigeration has made the 2, 1 and the fractional tonnage compressor practical.

The Direct Expansion System.—The very small compressor is designed so that it may be operated with the least possible attention. Such a compressor is always electric motor driven, the motor being either the induction or the repulsion-induction type as a rule. Automatic operation means that the compressor will not operate continuously, but intermittently with the demands of the load. There would be required in the automatic plant then:

a. A temperature control, set for the average temperature of the brine to be cooled, or of the air in the cold storage rooms. This is usually secured by means of a *thermostat* connected electrically with the necessary relays or devices to start or to stop the motor.

b. A suction pressure regulator, designed so as to maintain a constant suction pressure and therefore a constant boiling temperature of the refrigerant in the evaporating coils. In design this regulator is practically a pressure reducing valve.

c. A water control valve, so as to regulate the condensing water to periods of operation only, unless the condenser is air cooled.

In addition to the required valves already mentioned for the complete automatic machine there must be certain special features in order to safeguard against trouble. These include the excess pressure relief valve, and frequently a liquid valve to prevent the flooding of the evaporator coils during the period of shut down of the compressor. The compressor needs to be automatically lubricated, with special provision against the matter of *oil pumping*, and the shaft packing (such a machine is always of the vertical enclosed type if it is of the usual reciprocating type of design) must be tight against leakage of the refrigerant and yet not subject to much friction. The compressor should be provided with an equalizer connection so that on starting the compressor the pressure on the two sides of the cylinder will be the same, thus permitting the motor to get up to speed before full resistance to operation will take effect and thereby assisting in the decrease of electric motor troubles.

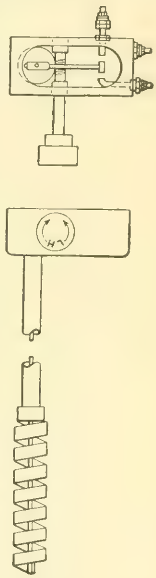


Fig. 102.—Englehart Thermostat.

The Thermostat.—The thermostat may be of the dissimilar metal type (Figs. 102 and 103) which depends on the variable expansion and contraction of the two metals used with the temperature change of the fluid to which it is exposed or it may be designed to utilize the effect of a volatile liquid exposed to a variable temperature. In the first example, the change of temperature causes a motion of the contact point which either makes contact with the *start* or the *stop* coils of the solenoid control.[1] This thermostat is placed directly in the fluid to be regulated in temperature. There are three different designs:

a. Two flat strips intimately joined together are coiled into a helix, one end of which is attached to a central rod and is free to turn (Fig. 102). The twisting moment of the rod is caused by the expansion and contraction of the coil with rise and fall of the temperature.

b. A bi-metallic thermo element in the form of a spiral (Fig. 103) forces a blade supported on knife edges so as to make contact at either the right or the left of the vertical, thereby energizing the necessary coils causing the motor to start or to stop.

[1] If the motor is properly designed with the necessary starting torque and the compressor is provided with a pressure equalizer, the compressor will be controlled satisfactorily.

c. The sylphon type of thermostat contains a liquid of the proper boiling temperature so as to secure the required differential pressure with a small change of temperature. The bulb (Figs. 104 and 105) may be placed in the cold room or the brine and a flexible tube connection may be made with the bellows. The mercoid control shown in the illustrations can be adjusted to make and break contact with a differential temperature of from 3 to 4 deg. F. The action of the sylphon is to tilt the mercury tube. The tube contains mercury and an inert gas

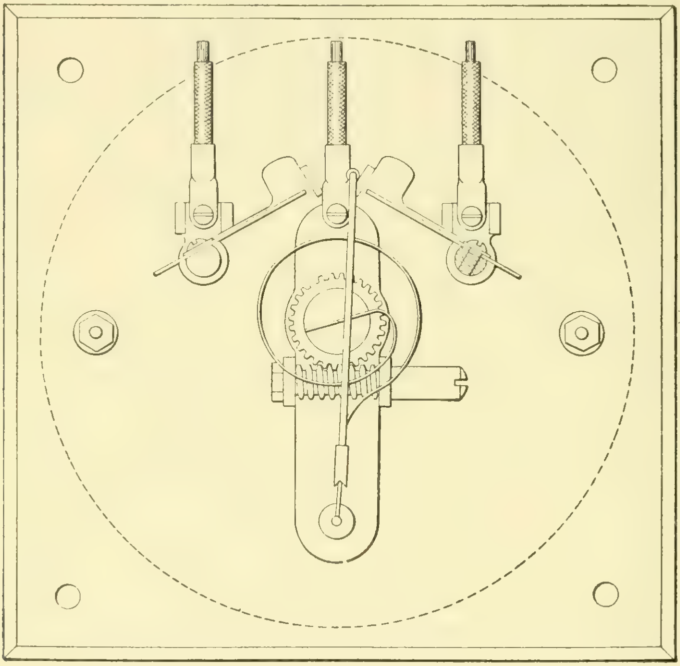


FIG. 103.—Monogram Full-line Voltage Thermostat.

to stifle the electric arc on *breaking*. The normal current is 10 amperes at 110 volts and 5 amperes at 220 volts on either alternating or direct current. This form of thermostat is very successful, as the bellows and the mercury tube have been tested to over one-half a million operations without signs of failure.

The Automatic Expansion Valve.—As previously mentioned, the automatic expansion valve is simply a pressure reducing valve. In the usual design it is of the diaphragm type (Figs. 105 to 110), whereby the movement of the diaphragm opens and closes the passage in the valve and permits a flow of the liquid refrigerant from the high-pressure side

to the evaporating coils. One side of the diaphragm is subject to the reduced pressure of the refrigerant, while the other side has the pressure

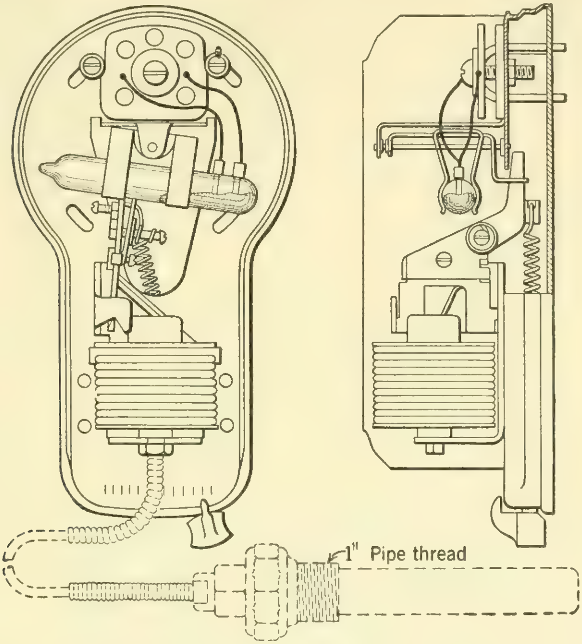


FIG. 104.—Mercoid Type Thermostat.

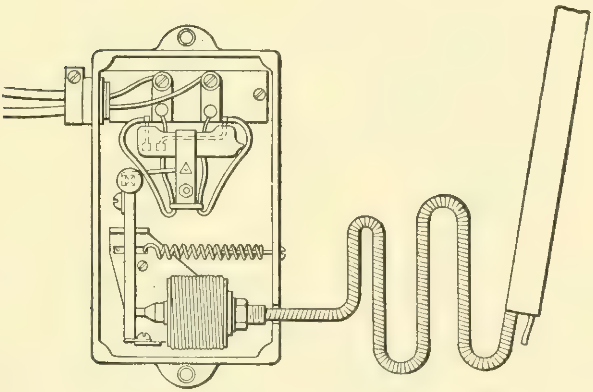


FIG. 105.—Absolute Contactor Thermostat.

exerted by a helical coil spring. This spring has a means of adjustment of the tension making it possible to obtain a control of the reduced (the suction) pressure over the usual operating range.

Such a valve as has just been described of necessity resembles the standard manual control expansion valve. In each there must be the same general design of piston seat and seat disc with provision for minimum wear and with a conical seat so that close adjustment of the flow of the refrigerant may be secured. In one design the piston seat and the seat disc are made of 30 per cent nickel steel, the diaphragm is made of monel metal and the body of the valve of close-grained cast iron. The Cash valve (Fig. 109) and the Peerless valve (Fig. 110) are examples of the construction just described. It is very essential that a scale trap be made an integral part of the valve or that one be placed adjacent to the valve and on the condenser side of it in order to prevent mill scale, sand, etc., from lodging on the valve seat. The Brunswick - Kroeschell valve (Fig. 111) obtains a sliding action which tends to free itself of any scale, and it is besides designed for complete closing as soon as the suction pressure increases above the amount at which the valve is set, thereby tending to prevent the flooding of the evaporating coils with liquid refrigerant during periods of shutdown of the compressor.

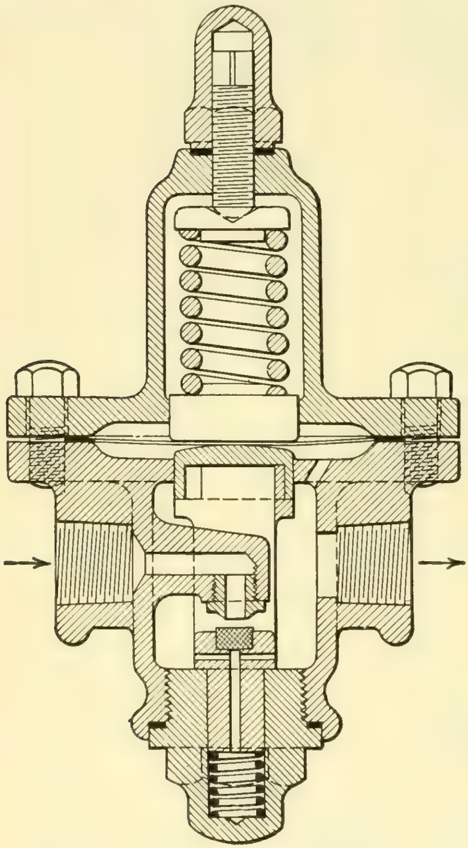


Fig. 106.—The Foster Expansion Valve.

The general criticism of automatic expansion valves is that they are likely to stick on account of the formation of a cold gummy material or ice on the the valve seats which prevents them from functioning properly. The Climax valve (Fig. 112) is entirely different in design from those already described. The diaphragm is replaced by a sylphon subject to the evaporator temperature. Change of temperature causes a breathing of the sylphon which causes a lever (the free end of which is

continually oscillating) to move axially. For certain positions of the lever engagement is made with a pin which causes the valve to open and close with great rapidity while at other positions of the cam the valve will remain closed entirely. It is evident that the suction pressure controls the amount of opening of the valve, and that no sticking of the valve can be possible. There is the objection, however, that the expansion valve must be attached to the compressor.

Water Control.—The automatic compressor requires a control on the water, both as regards the amount of the water used and the adjust-

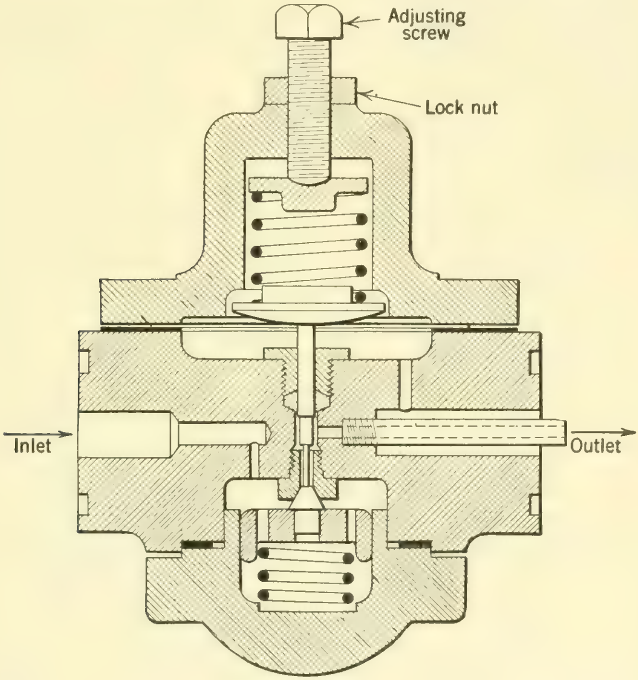


Fig. 107.—Alco Expansion Valve.

ment of the water to the actual periods of operation. There are a number of different designs. Some machines have used direct connection to the shaft of the compressor so that when the compressor is in operation the condensing water will flow, but this form of water valve gives no control of the amount of water being used. Another type (Fig. 114) makes use of a sylphon with the valve placed at the exit from the condenser, so arranged as to have a slight flow even at periods of shutdown. The temperature of the water passing the sylphon affects the amount of the opening through which the water

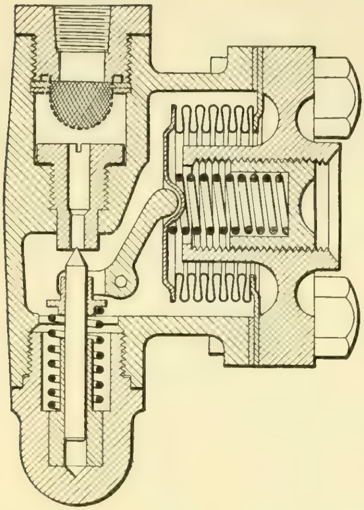


FIG. 108.—American Radiator Expansion Valve.

may pass and thereby permits a greater or lesser amount of water to pass, depending on the initial temperature of the water and the load on the condenser.

The more common type of water regulator valve is shown in Fig. 115 (the Cash valve) and in Fig. 116 (the Monogram valve). In each of these the pressure of the refrigerant is exerted on a diaphragm loaded on the upper part by means of a helical spring which is set for some particular unit pressure, for example, 150 lb. per square inch. When the compressor starts up the condenser pressure increases until it gets to an amount corresponding to that at which the valve was set, whereupon the valve will open and water will begin to flow. The amount of water passing through the valve, and through the condenser, will increase with an increase of pressure until the capacity of the valve is reached. Some adjustment for the individual installation can be obtained by varying the relative position of the spring yoke through the use of the nuts on the standard or by varying the length of the valve stem.

An excess pressure, caused by a failure of the condenser water supply, can be safeguarded against by a pop-relief valve, but such a method is not usual in automatic refrigerating machines, even to discharge the gas into the low-pressure side of the system. A better method of

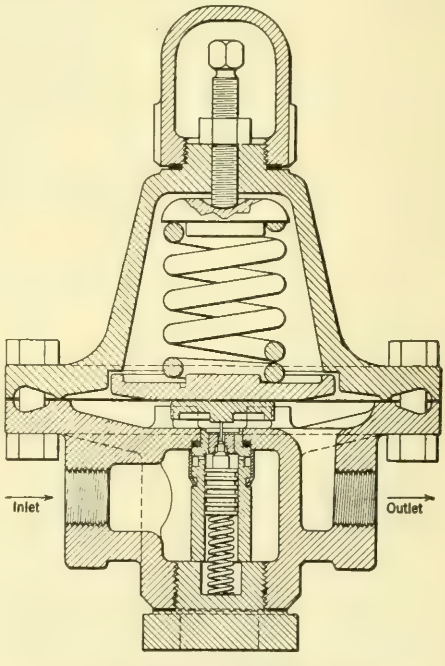


FIG. 109.—Cash Expansion Valve.

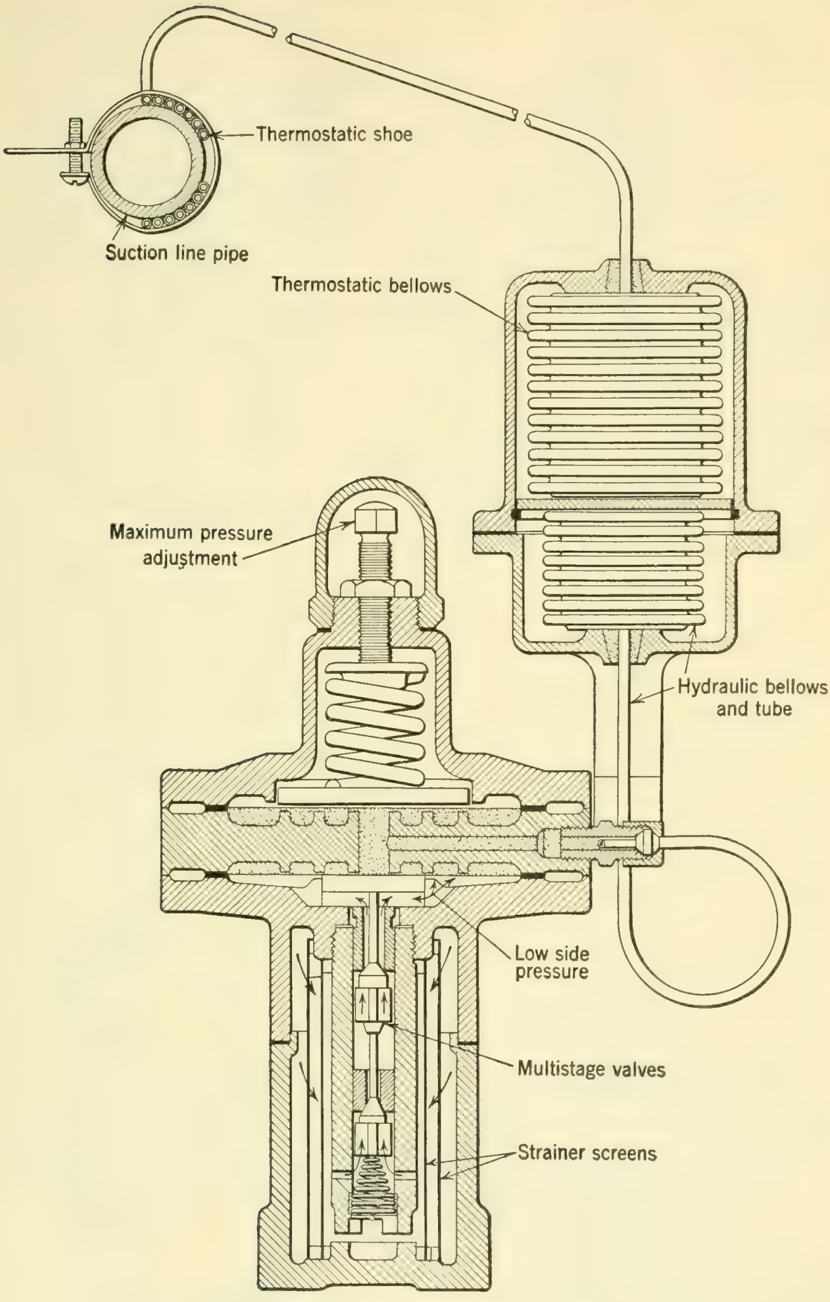


FIG. 110.—Peerless Economizer Valve.

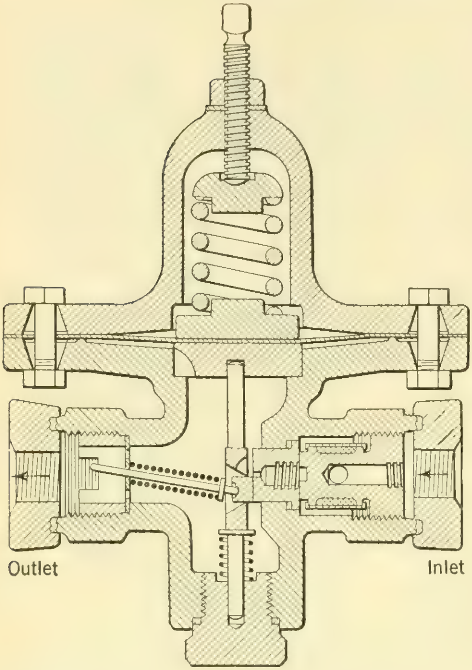


FIG. 111. — Brunswick-Kroeschell Expansion Valve.

control for small machines is to stop the compressor until such time as the pressure decreases to the allowable working amount. In larger machines we may have a condenser pressure circuit breaker, which has to be controlled manually.

Figure 117 shows one type of water pressure failure switch. The water pressure is exerted on one side of the rubber diaphragm, the other side of which has the pressure of a light coil spring. The valve stem motion is communicated by means of a toggle joint to the electric connection, which can be of the knife-edge switch design.

Consideration up to the present has been given only to those plants where there is but one set of evaporating coils. The problem is much complicated by having two or more such coils, the loads of which are variable and entirely independent of one another. For such an automatic plant, it is evident that the compressor cannot be shut down until the conditions in all the evaporating coils are satisfied. The thermostatic control to each room, can not be attached to the compressor but only to the

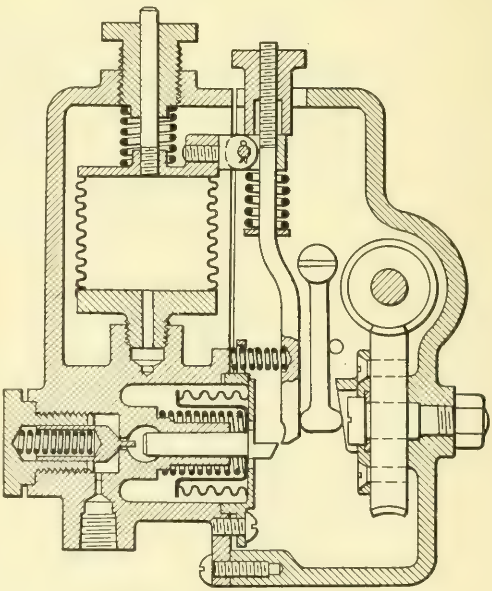


FIG. 112.—The Climax Expansion Valve.

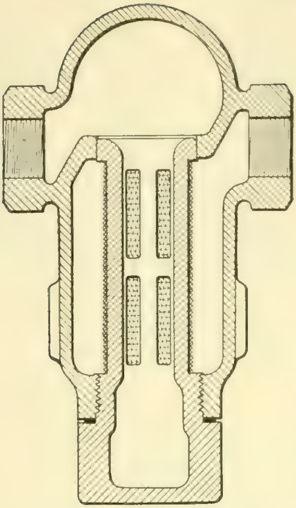


FIG. 113.—Cash Expansion Valve Strainer.

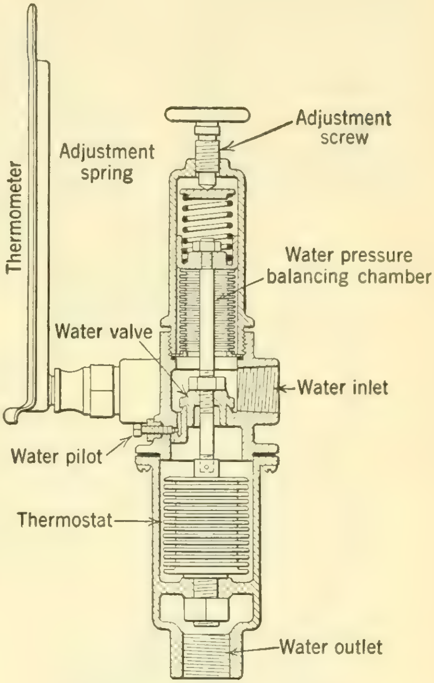


FIG. 114.—Water Regulator.

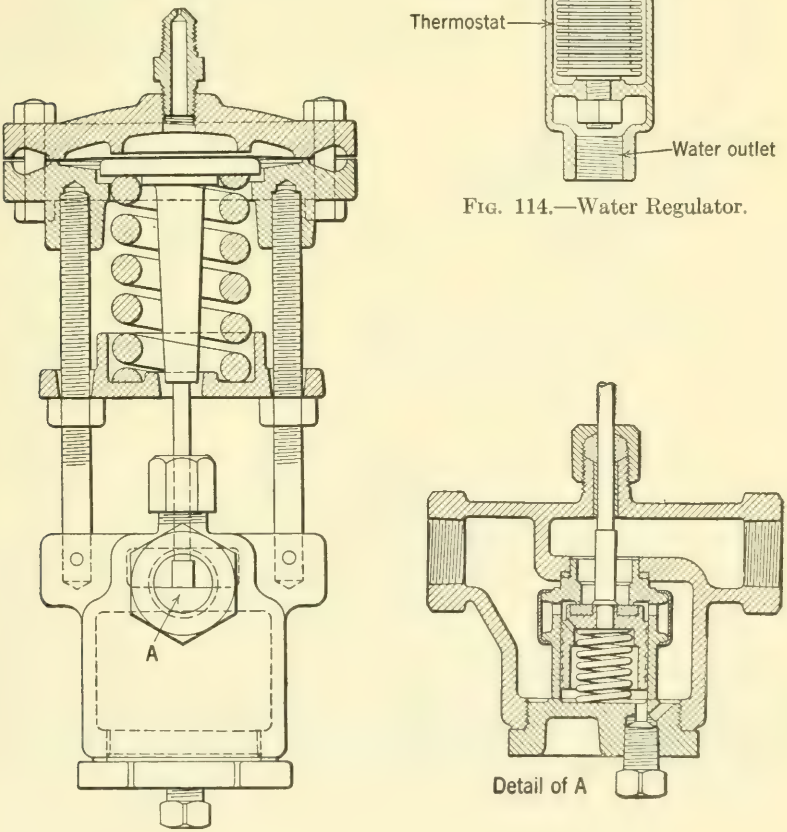


FIG. 115.—Cash Water Regulator.

controls to the individual piping. The compressor operation now will need to be obtained, not from the temperature of the rooms, but from the drop of the suction pressure in consequence of stopping the liquid feed to *all the evaporator coils.*

Figure 118 shows a Monogram type of magnetic stop valve for the regulation of the flow of liquid to a set of evaporator coils (in conjunction with an automatic expansion valve). The thermostat energizes the relay and through the relay the magnet operates the toggle joint.

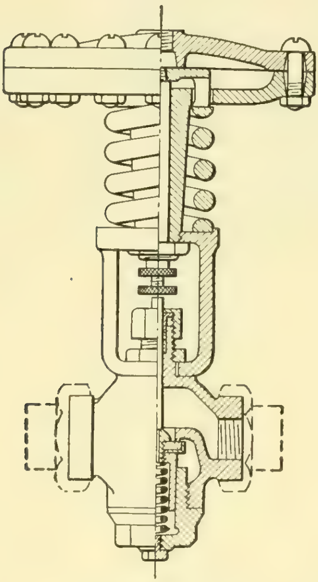


FIG. 116.—Monogram Water Regulator.

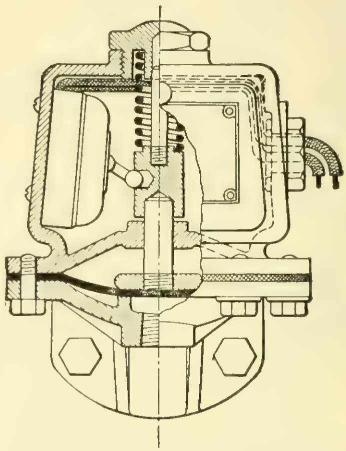


FIG. 117.—Water Pressure Failure Switch.

As soon as the liquid flow is stopped by the closing of the liquid valve the absorption of heat by that particular evaporator coil will stop as quickly as the liquid refrigerant in the coal is vaporized.

Figure 119 shows a form of an ammonia pressure-actuated switch to be used on the suction side in order to control the operation of the compressor where several evaporating coils are used. This valve can be set to operate at some pressure lower than that required during the operation—for example at 5-lb. gage. Such a pressure could not be obtained with any of the evaporator coils in operation if the expansion valves are set at 15 or 20 lb. gage. This suction pressure could only be obtained when all the coils are shut off from the liquid refrigerant,

at which time the compressor will function only to pump down the low pressure system. Figure 120 shows a multiple thermostatic control for

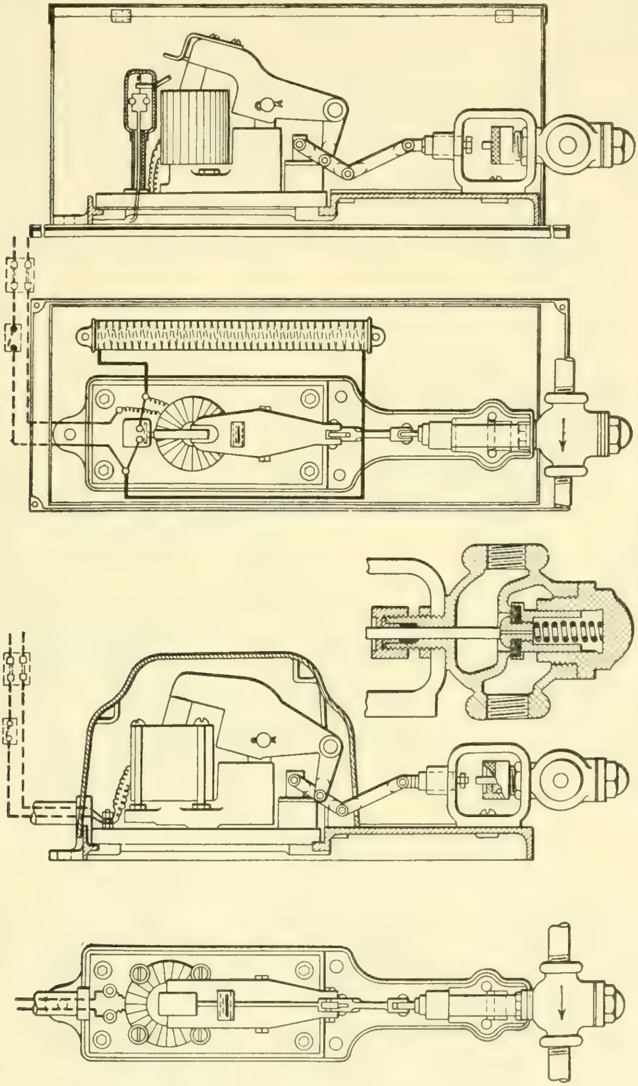


Fig. 118.—Magnetic Stop Valve.

two or more evaporating coils using the controls already described and illustrated.

Somewhat different from what has just been described is the Hilger No-freeze (Fig. 123) back valve. In essentials the liquid, after leaving

the automatic expansion valve, passes through the upper part of the no-freeze back valve. The suction gas on its return has to pass through the lower part of the valve which is subject to the suction gas temperature in such a way[2] as to operate a valve in the liquid line, even to the point of complete closure.

Dual Thermostats.—In laying the piping for automatic plants some very interesting arrangements are made at times. An example of such a design might be the arrangement prepared for the florist who has a

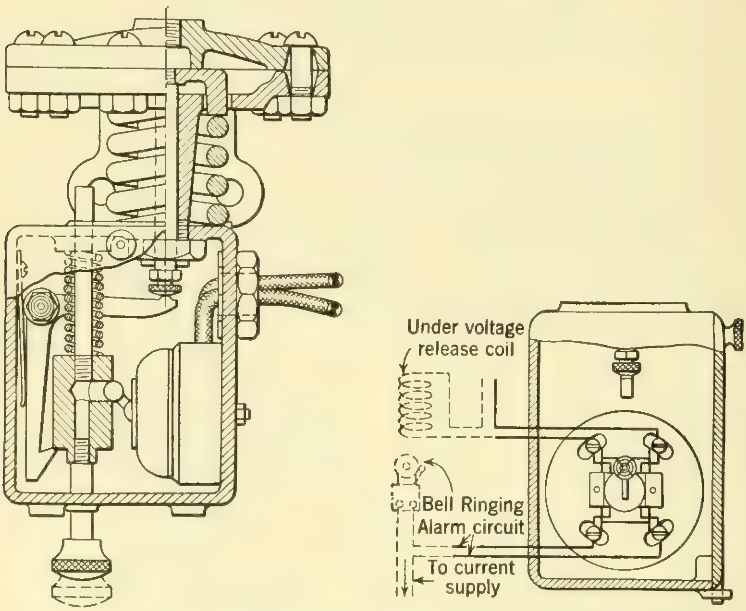


FIG. 119.—Ammonia Pressure-actuated Switch for Maximum Pressure.

show case and a cold storage room, with the piping connected in series, with a thermostat in each. The liquid enters the coils in the show case, and on leaving these coils passes to the piping in the cold storage room. The show case has a thermostat set at 45 deg. F., and the cold storage room has one set at 35 or 36 deg. F. The second thermostat acts only to protect the flowers in the cold storage room from freezing should an extreme condition develop where the cold storage room is cooled to a dangerous temperature before the first thermostat has a chance to act.

[2] The decreased temperature of the ammonia gas contained in the center of the valve exerts a decreased pressure on the diaphragm and the diaphragm is pressed downward by the fluid pressure on the upper part.

Brine Circulation.—Where brine is circulated the automatic features are relatively simple. Such a plant would have brine storage, and in the simple arrangement the thermostat would start or stop the compressor for a certain brine temperature range. Such a control is obtained easily through the mercoid or other form of thermostat switch. Where the brine system consists of a plurality of coils, each of which requires a control, there will need to be a thermostatically controlled valve on each coil equipped with solenoids in order to give the proper opening and closing required (Fig. 124).

The Small Refrigerating Machine.—The small refrigerating machine is essentially automatic. It is inconceivable that the fractional tonnage machine, used now so extensively for ice cream cabinets, small boxes, and for the household refrigerators, could be operated in any other manner. To a lesser extent, perhaps, the same thing is true of the 1, 2, 4, 5, and even the 10-ton machine as the demand is increasing for a machine in these sizes which does not require constant attention from an expert operator. It is appropriate, therefore, to

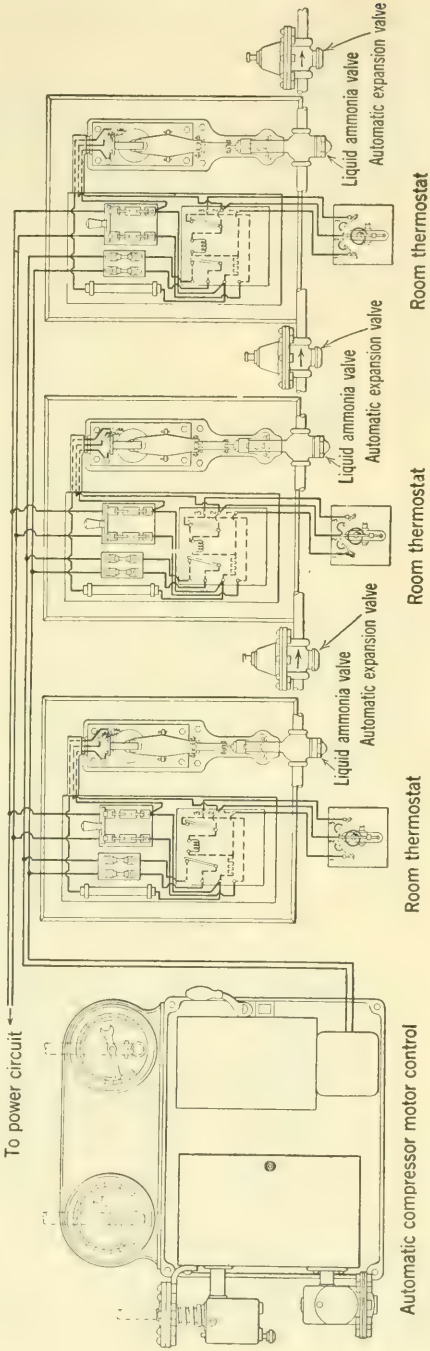


FIG. 120.—Method of Automatic Control.

take up these small machines an an example of the automatic features just described. In all probability no other type of mechanical refrig-

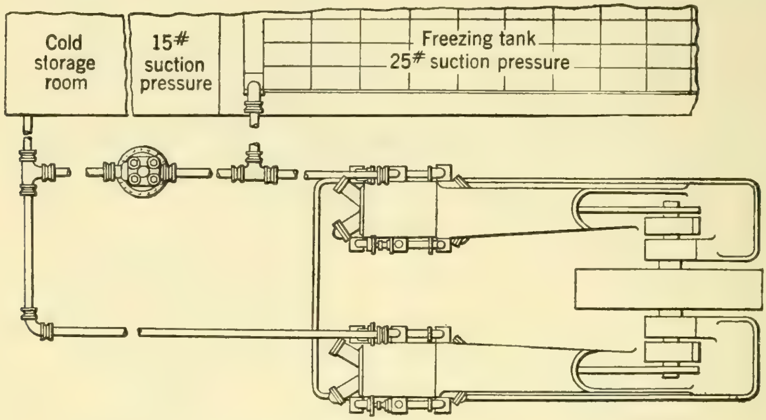


FIG. 121*a*.

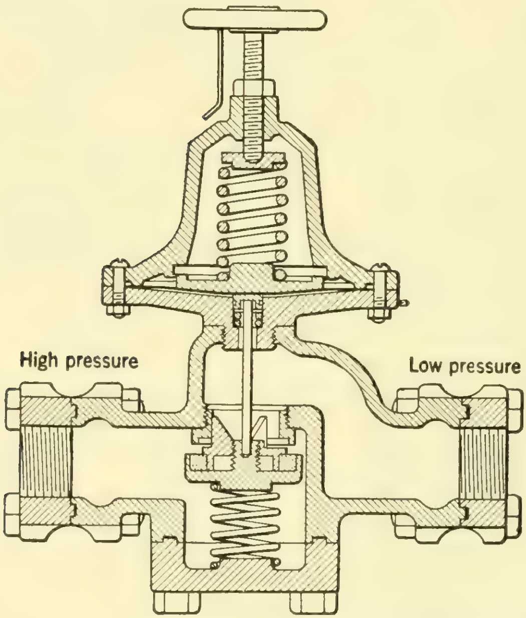


FIG. 121*b*.

FIG. 121.—Ammonia Pressure Regulator.

eration is attracting the interest that the small machine does, and its popularity derives from the following reasons:

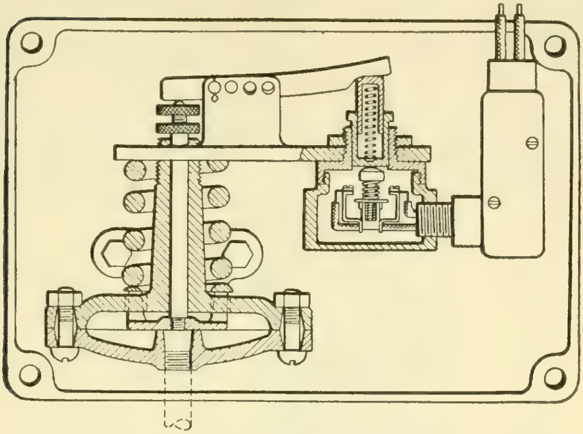


FIG. 122.—Pressure-actuated Switch.

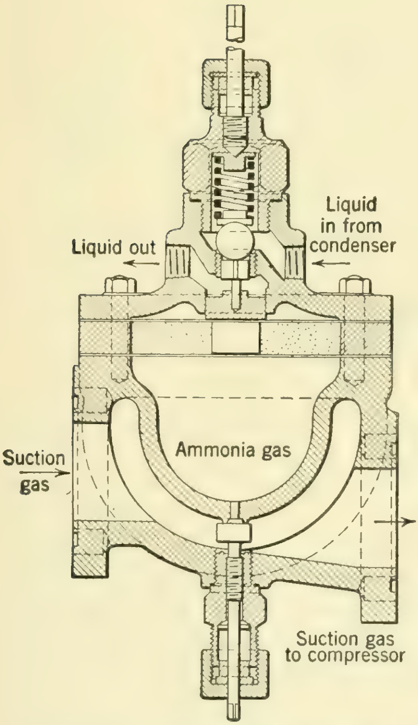


FIG. 123.—Hilger Non-Freeze Back Valve.

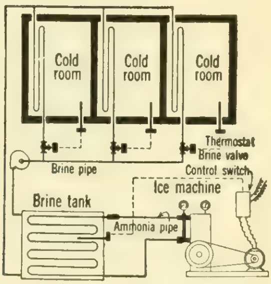


FIG. 124.—Thermostatic Control of Brine.

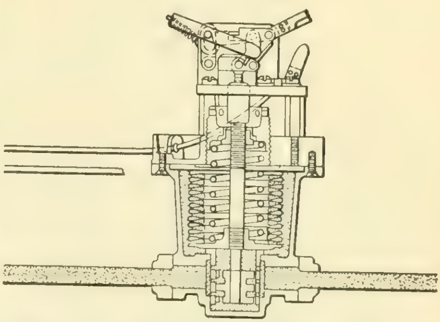


FIG. 125.—Pressure Control on Household Compressor.

Ice for small cold storage boxes or refrigerators is frequently found to be dirty and unreliable. At times the ice delivery service has not been of the best, and some localities have not been able to secure ice at all. Besides this, the temperature in the boxes—for storage without the use of salt on the ice—has been found to be unsatisfactory in many cases. The refrigerators may be designed poorly or be in poor condition, with the result[3] that temperatures are not lower than 45 to 55 deg. F.

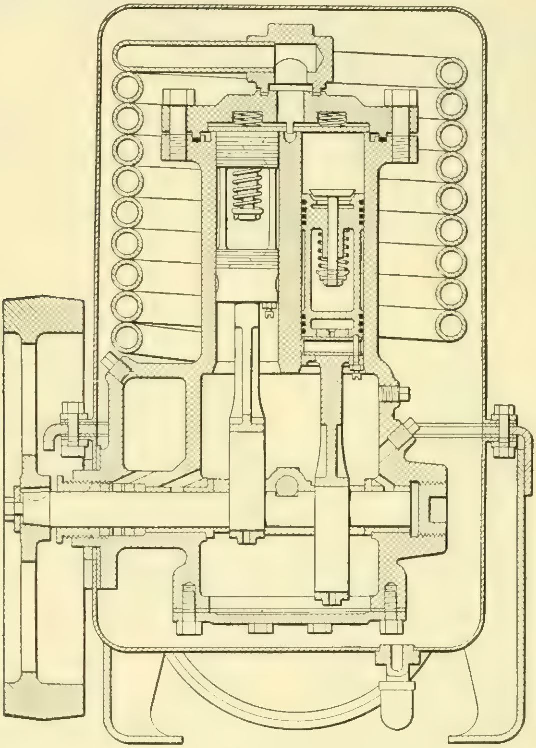


FIG. 126.—One-half to One-quarter-ton Compressor.

In general, it can be said that temperatures in such a box cannot be obtained lower than 35 and are more often greater than 40 deg. F

The mechanically cooled refrigerator gives practically any temperature desired, controlled thermostatically within a slight range. At times it gives a drier box, although good circulation will do the same

[3] The Transactions N. E. L. A., 1923, give results of tests by Prof. A. M. Greene Jr., in which it was found that only 4 per cent of 250 ice-cooled refrigerators which were tested operated with a temperature less than 45 deg. F.

for the ice-cooled refrigerator. The variations in the temperature at different times of the day are very slight as compared with ice cooling, for the latter requires a full supply of ice in the ice compartment at all times in order to get good results.

As regards costs of operation, the real household refrigerating machine cannot compete with ice, if ice costs are those usually found in the larger cities. The operation cost, which must include the cost of power, water (if used), interest and depreciation on the mechanical equipment, repairs, etc., is usually an appreciable amount, and frequently will be found to be about twice the cost of the operation of the

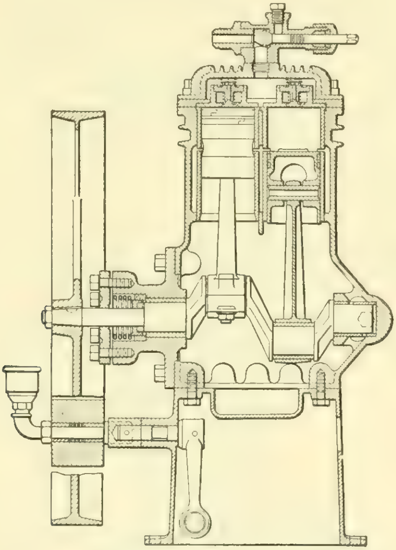


FIG. 127.—Kelvinator Compressor.

ice-cooled refrigerator, especially in those localities where ice does not have to be used for the entire 12 months. There are a number of automatic features in the household refrigerating machine; the thermostat, the excess pressure cut-out, and the suction pressure control. If the condenser is water cooled, the machine should have some means of regulation of this water. The companies selling service during the first year of operation consider that they are lucky if an average of only 2.5 calls per year per machine are required. This service is required to adjust the automatic features as well as to attend to the more serious troubles due to the loss of the refrigerant, loss of oil, or to failure of the expansion valve to operate satisfactorily. Most concerns selling these small machines provide free service for the first year of

operation, and after the first year render service at about $1 per month. Without question these small compressors are becoming more perfect every year, and their popularity is increasing rapidly in spite of the undoubted extra cost. This statement does not apply to the $\frac{1}{6}$-ton machines which are used to a large extent for ice cream cabinets where temperatures of 10 deg. F. or lower are required.

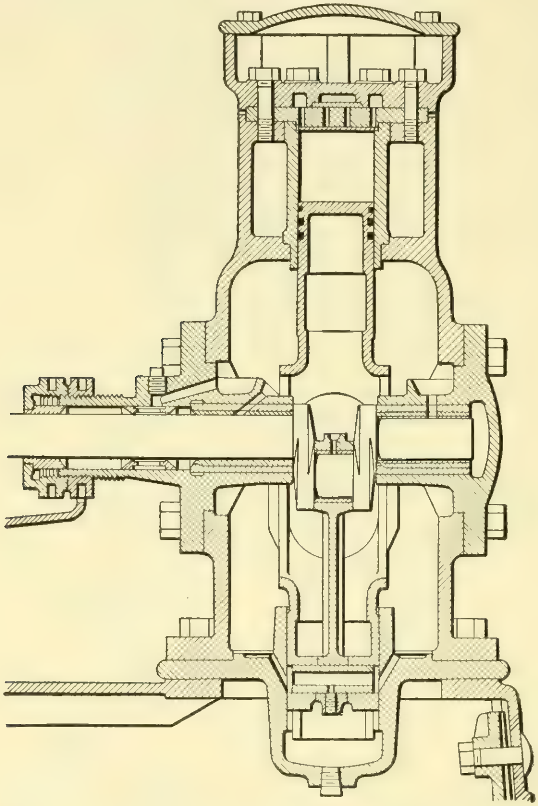


FIG. 128.—Universal Compressor.

The Refrigerant and the Operating Conditions.—The usual refrigerants are ammonia, sulphur dioxide, ethyl and methyl chloride, butane, isolutane, propane, etc. Carbon dioxide is not used as a refrigerant for household compressors because of its excessive pressure. By means of the automatic suction pressure control and the use of a suitable thermostat, the temperature carried in the box can be set at will, and is usually set at 35 to 40 deg. F., with a variation of about 2 degrees on either side of the desired temperature. With the exception of the absorption

machine, the household refrigerating machine is always electrically driven, using a $\frac{1}{4}$-hp. motor, and the power consumption is about 1.5 kw.-hr. per 24 hours. The actual running time is from 5 to 9 hours, and the compressor starts and stops approximately 6 to 10 times per 24 hours. At present there are some air cooled, up to about

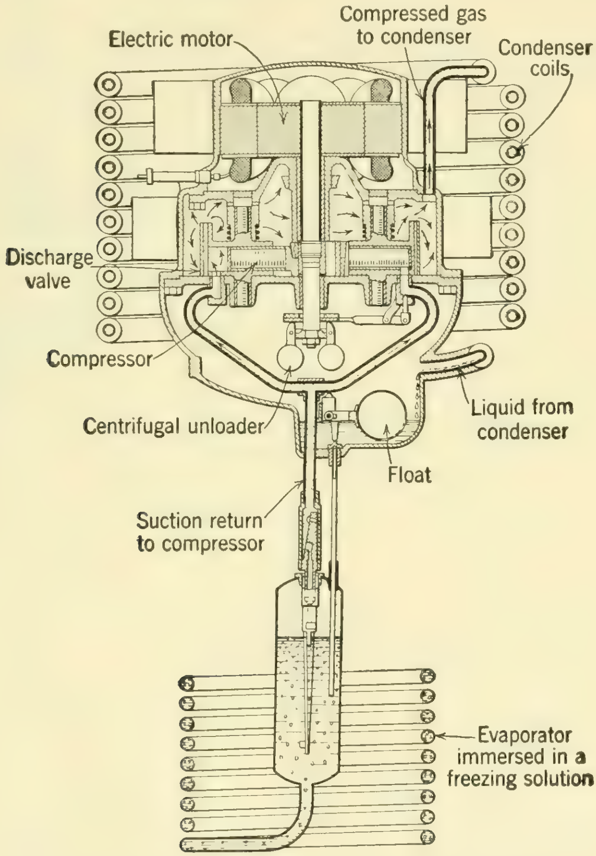


FIG. 129.—The General Electric Compressor.

$\frac{1}{2}$-ton capacity, and some water-cooled condensers. The air-cooled condenser has a higher temperature of liquefaction of the refrigerant with a temperature of liquefaction of the working medium some 10 to 20 deg. higher than the room air temperature, and therefore more power is required of the compressor than with a similar case using tap water over the condenser. In addition, the air-cooled condenser requires a fan to drive the air through the condenser, and this is an added load to

the motor, but usually not exceeding 0.02 hp. on the $\frac{1}{4}$-ton size. In the water-cooled condenser the water consumed varies from 10 to 15 cu. ft.

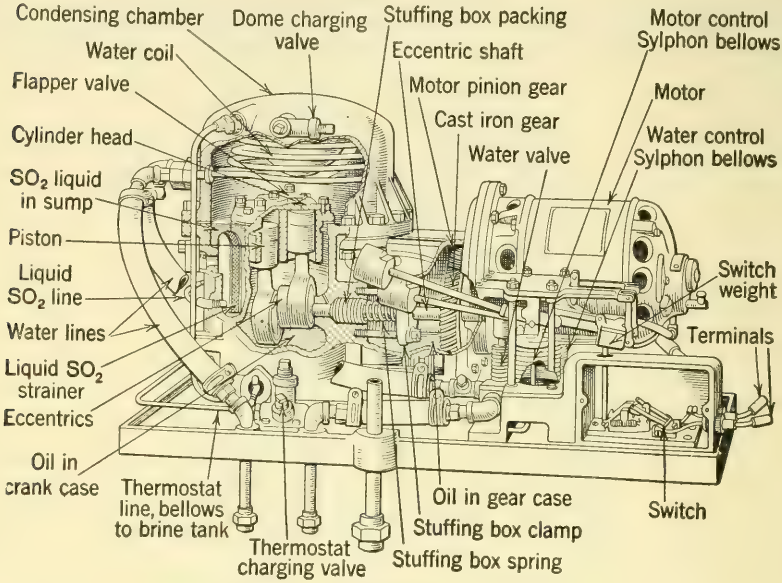


Fig. 130.—The Frigidaire Compressor.

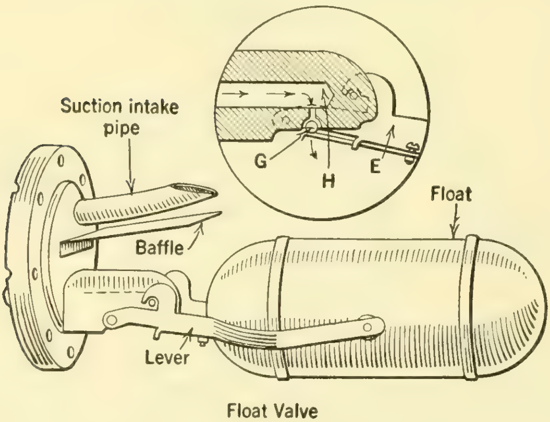


Fig. 131.—The Float Valve for Household Compressors.

per day. However, the air-cooled condenser has a great advantage in being simpler in design, is not dependent on the continuity of the water supply, and appears to be more and more popular.

The expansion valve may be of two principal designs—the diaphragm type (Fig. 125) and the float type (Fig. 131). In the household machine the low-pressure side is relatively small and the charge of the refrigerant is relatively quite nominal. This is the reason why the float type of valve can be used and in the sulphur dioxide compressor this kind of expansion valve works out best as regards the lubrication of the compressor. The diaphragm type of compressor can be set for any pressure, and this is usually at such an amount as will give a brine temperature of from 10 to 20 deg. F. The main objection to the diaphragm type of

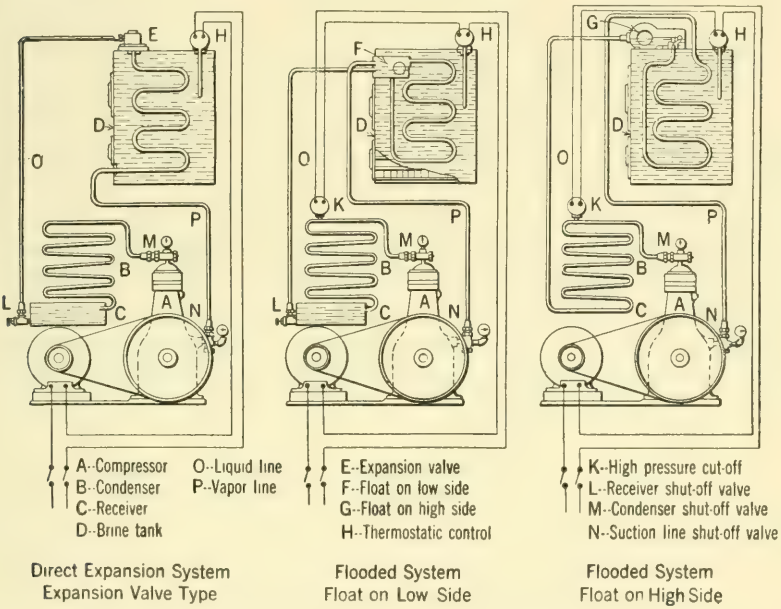


FIG. 132.—Typical Piping Arrangements—Household Compressors.

valve is that when the compressor is stopped the valve frequently will permit the flooding of the evaporating coils, creating a condition that will require some time to pump down after the compressor is started again, thus resembling the action of the float type of valve.

The thermostat is usually designed with the sylphon containing a volatile liquid-like sulphur dioxide. As the temperature rises this liquid exerts a higher pressure and the the design is such that the increment increase of the pressure will operate the sylphon which operates the control switch. In this manner the motor may be started or stopped. If the condenser is water cooled, this water may be controlled in a number of ways. A sylphon may be attached to the water outlet, in

such a manner that the temperature of the exit water will vary the amount of the water flowing to suit the conditions of the plant, or the water control may be attached directly to the compressor. Still another method is to use a diaphragm type of valve, operated by the condenser pressure. Figures 126 to 130 show typical designs of household compressors, several of which are no longer being built. Figure 132 gives three types of arrangements in common use.

The household compressor or absorption machine has to be designed so as to be very self-contained. In the compressor the design must be such that the oil will not be lost from the system as this would produce a loss of lubrication; nor must there be any loss of the refrigerant to the atmosphere, resulting in a failure of the machine to function, and, in the case of the sulphur dioxide machine, in the corrosion of that metal near where the leak takes place. The automatic controls need to be so well designed that they will function for a period of time without adjustment or repair. Where gas only is available the absorption machine may work in well. This is of the intermittent type, wherein the heating of the generator by the gas flame (or electric resistance) boils out the ammonia gas which is condensed and stored in the liquid receiver. After the proper time-interval the flame is shut off and water is passed through the generator, which is now functioning as an absorber, and the liquid after boiling in the expansion coils returns to the generator again in order to re-enter solution.

CHAPTER VI

HEAT TRANSFER

Refrigeration, like other branches of heat engineering, is vitally interested in heat transfer, both from the viewpoint of heat leakage through the building material and from the consideration of the flow of heat through the various parts of the refrigerating cycle. At times the refrigerating load is entirely one of heat leakage, and so the insulation against this leakage is one of the primary subjects for careful thought. In the refrigerating cycle heat transfer is all important in the condenser, brine cooler, or other evaporating surfaces of the low-pressure side, and in the generator, rectifier, exchanger, absorber, etc., of the absorption machine. The problem in many cases is also complicated by the variation in the velocity of the fluid in contact with the surface and by the insulating effect of the air, oil, etc., to be found inside the pipes in nearly every case. Some of these factors will be taken up briefly.

Theory of Heat Transmission.—In general, heat transfer may take place in three different ways: by radiation, by conduction, and by convection. In refrigeration, however, the temperature range is small, and it is seldom that the effect of radiation is appreciable.

Radiation.—Every surface has a definite emitting power and a definite absorbing power for radiant heat. This radiant heat is transmitted from one body to another by means of ether waves, so that the presence of air (or the lack of it) does not affect radiation materially. The amount of heat transmitted varies as the *fourth* power of the absolute temperature for black bodies. This relation has been shown by Stefan and Boltzmann, expressed in English units, by

$$Q_{\text{radiation}} = RA(T^4{}_2 - T^4{}_1)$$

where

T_2 and T_1 are the surface temperatures in degrees absolute of the bodies in question,

A is the area in square feet,

R is a constant.

In refrigeration, as radiation by itself plays such a small part, it can be neglected in most cases. Sometimes, however, in reading temperatures, the thermometer may give incorrect readings when exposed to direct radiation from hotter or colder bodies because of this factor.

Convection.—Convection losses have to do with a fluid in motion past the surface in question. It is worthy of remembrance, however, that a layer of the fluid in contact with a solid is always at rest because of adhesion, and a finite thickness of the fluid adjacent to the solid—due to friction and the viscosity of the fluid—is always either at rest or is moving too slowly to have any appreciable effect on the resistance to the passage of heat. If the fluid happens to be oil, it appears that the surface film, which is at rest, has the same resistance to the flow of heat as cork-board of the same thickness. The thickness of this film is dependent on the viscosity and the density of the fluid and the velocity of the current next to the stagnant layer. It varies, with air, from nearly 0 to $\frac{1}{2}$ in. Obviously, the value of this skin resistance is not of appreciable importance in surfaces using cork or other heat insulation, but for glass it is a very important factor, although even glass is not used in mechanical refrigeration except in multiple thicknesses and in very special cases.

Peclet's laws for convection, expressed in English units, are given by

$$Q_{\text{convection}} = NV^{-n}A(t_2 - t_1)$$

where

N is a coefficient of convection and a variable,

V^{-n} is some power of the velocity in feet per second across the section,

$(t_2 - t_1)$ is the difference in temperature between the surface and the surrounding air.

Conduction.—Conduction is the means whereby heat is conveyed through molecular vibration, under which conception there is established a temperature gradient in the material. It is well established experimentally that if the material is homogeneous, and nominal temperatures are being employed that the heat flow becomes:

$$Q_{\text{conduction in B.t.u. per hour}} = Ak(t_2 - t_1)$$

Where

A is the area in square feet,

k is the coefficient of heat transfer in B.t.u. per square foot per degree difference per hour, and

$(t_2 - t_1)$ is the temperature difference on the two sides of the heat transfer surface in deg. F.

The effect of radiation and convection can be approximated by the method used by Willard and Lichty[1] by means of the following:

If the temperature differences are small, as they are in refrigerating practice, the value of the radiation and convection losses per square foot of surface per hour becomes:

$$Q_{\text{radiation}} + Q_{\text{convection}} = R[(t + 460) - (t_1 + 460)] + NV^{-n}(t - t_1)$$

and for walls small in height, and in still air this becomes

$$Q_{\text{radiation}} + Q_{\text{convection}} = k_1(t - t_1),$$

but where an outside surface is exposed to nominal air velocity this becomes:

$$Q_{\text{radiation}} + Q_{\text{convection}} = k_2(t_2 - t_0)$$

and

$$Q_{\text{conduction}} = \frac{C}{T}(t_1 - t_2)$$

where

T = the thickness in inches, and

C = is the conductivity of the material.

As Q at any part of the wall must be equal to that of any other part of the section,

$$Q = k_1(t - t_1) = \frac{C}{T}(t_1 - t_2) = k_2(t_2 - t_0) = U(t - t_0).$$

Therefore,

$$\frac{1}{k_1} + \frac{T}{C} + \frac{1}{k_2} = \frac{t - t_1}{Q} + \frac{t_1 - t_2}{Q} + \frac{t_2 - t_0}{Q} = \frac{t - t_0}{Q} = \frac{1}{U}$$

and

$$U = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2} + \frac{T}{C}} + \quad \text{etc.}$$

It is clear from the foregoing that if a compound wall had been considered in the formula it would have become

$$U = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2} + \frac{T_1}{C_1} + \frac{T_2}{C_2} + \frac{T_3}{C_3}} \quad \text{etc.}$$

This equation is the one generally used in the computation of the heat leakage in buildings, cold storage rooms and boxes, but in order to use this formula the values for k_1 and k_2 must be known as well as the

[1] Willard and Lichty, Bulletin No. 102 of the Engineering Experiment Station, University of Illinois.

average values of C for building material and insulation. The values for k_1 are given in Table 46, and it was recommended by Willard and Lichty that the values for k_2 be taken as three times those for k_1 for a nominal velocity of the wind of not more than 15 miles per hour.

Values[2] of C for insulation are given in Table 43 and for building material are given in Table 44. For convenience Table 45 is also included, calculated for 1.0 degree difference in temperature, and for the heat transfer per square foot per hour for 20, 40, 60, 80 and 100 degrees difference in temperature on the two sides of the built-up material. In these calculations the values of k_1 and k_2 are included,

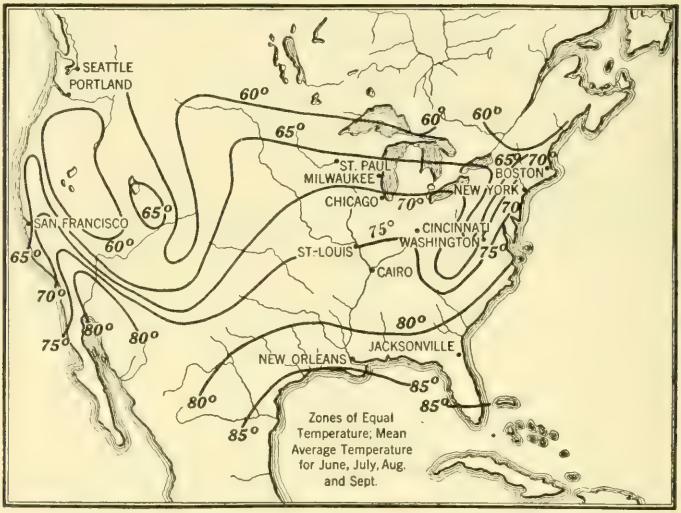


FIG. 133.—Zones of Average Temperature for July, August and September.

although it is very doubtful whether such refinements are justified, especially as the personal equation of the workmen requires that a factor of about 25 per cent be added to the heat transfer as calculated. As a rule the skin effect increases the value of the denominator by 0.77 for each *inside*, and by 0.25 for each *outside* surface. In order to get an idea of the average temperature of the air throughout the United States, average isothermals for the three summer months are given in Fig. 133. In cold storage work it is not the maximum air temperature that should be taken for the calculation of the heat leakage, but the maximum average sustained temperature for three or more days.

[2] A very complete review of insulation values is given in the Report of the Insulation Committee, American Society of Refrigerating Engineers, 1924.

TABLE 43

THERMAL CONDUCTIVITIES OF VARIOUS INSULATING MATERIALS—U. S. BUREAU OF STANDARDS, DEPARTMENT OF COMMERCE

Material	Conductivity, B.t.u. per Square Foot per Degree Difference of Temperature per Hour	Density	Nature of Material
Air	0.167	....	Horizontal layer heated from above.
Colorax	0.221	0.064	Fluffy, finely-divided mineral material.
Hair felt	0.246	0.27	Hairfelt confined between layers building paper.
Keystone hair	0.271	0.30	
Pure wool	0.242	0.107	Firmly packed.
Pure wool	0.242	0.102	Firmly packed
Pure wool	0.262	0.061	Loosely packed.
Pure wool	0.292	0.039	Very loosely packed.
Cotton wool	0.292	0.100	Firmly packed.
Insulite	0.296	0.19	Pressed wood pulp, rigid, fairly strong.
Corkboard (pure)	0.317	0.18	
Granulated cork	0.34	0.160	
Eelgrass	0.321	0.25	Inclosed in burlap.
Flaxlinum	0.329	0.18	Vegetable fibres, firm and flexible.
Fibrofelt	0.329	0.18	Vegetable fibres, firm and flexible.
Balsa wood	0.346	0.12	Very light and soft.
Rock cork	0.346	0.33	Pressed rock wool with binder, rigid.
Waterproof lith	0.409	0.27	Rock wool, vegetable binder, not flexed.
Pulp board	0.433	....	Stiff pasteboard.
Air cell, ½ in.	0.446	0.14	Corrugated asbestos paper enclosing air spaces.
Air cell, 1 in.	0.479	0.14	
Asbestos paper	0.491	0.50	Fairly firm but easily broken.
Infusorial earth	0.579	0.69	(Block).
Fire felt (sheets)	0.596	0.42	Asbestos sheets coated with cement, rigid.
Fire felt, roll	0.637	0.68	Soft, flexible asbestos.
3-ply Regal roofing	0.696	0.88	Flexible tar roofing.
Asbestos mill board	0.847	0.97	Pressed asbestos, fairly firm, easily broken.
Cypress	0.657	0.46	
White pine	0.792	0.50	
Virginia pine	0.957	0.55	
Oak	1.000	0.61	
Hard maple	1.125	0.71	
Asbestos wood	2.710	1.97	Asbestos and cement.

The insulation of pipes is given by Heilman [3] in Table 47 and the equation at the bottom of this table. However, the problem is slightly different from that of flat plates (as walls, floors, etc.), and it is usual to take the values of different insulations per one foot of length of nominal pipe size as found by experiment. The thicknesses used are those for drinking water, standard brine and extra thick brine. Values for heat transfer for these thicknesses are given in Tables 48 and 49.

TABLE 44

COEFFICIENTS BASED ON HEAT TRANSMISSION TESTS [B.t.u. per Hour]

(These values are based on the tests run under most satisfactory conditions)

Material	C per 1 In. Thickness per Square Foot per 1 Deg. F.	K Still Air per 1 Deg. F.
Brick wall (mortar bond and dry conditions)	4.00 (5.00)	1.40
Concrete, 1–2–4 mixture	8.30	1.30
Wood (fir, one surface finished)	1.00	1.40
Corkboard	0.32	1.25
Magnesia board	0.50	1.45
Glass (actual glass 91.4 per cent of total area)	2.06‡	1.5 to 2.00
2-in. tile, ½-in. plaster on both surfaces	1.00§	1.10
4-in. tile, ½-in. plaster on both surfaces	0.60§	1.10
6-in. tile, ½-in. plaster on both surfaces	0.47§	1.10
2-in. tile, plastered as above and roofing covered	0.84	1.25
Asbestos board	0.50	1.60
Sheet asbestos	0.30	1.40
Double glass, ½-in. air space (glass 69.3 per cent of total area)	1.50‡§	2.00
Roofing*	5.30§	1.25
Air space†	1.00–1.70§	
Mortar	8.0	0.93
Shavings	0.42	

* Calculated from values of C for 2-in. tile with and without roofing. $\left(\frac{1}{C} = \frac{1}{C_1} + \frac{1}{C_2}\right)$

† See "Air Spaces."

‡ Per square foot of actual glass set in wood frame but based on total heat transmitted.

§ For thickness and construction stated, not per 1 in. of thickness.

Kinds of Insulations.—Referring to Table 43, it will be seen that there are a large number of substances that are suitable for low-temperature insulation, but a choice of an insulating material should not be made without all pertinent factors entering into the consideration. For example, the value of C for ½-in. air space is given as 10.7 B.t.u., a

[3] R. H. Heilman, Mechanical Engineering, 1924.

1-in. air space is 11.5 and a 1-in. horizontal layer is given as 4 B.t.u. per hour. It is understood that air is the best insulator known, provided convection can be prevented, and it can be prevented best by the use of a material with a large number of air cells of small size separated from one another as is to be found in cork, hair felt, wood and vegetable material of various sorts, as flaxlinum, fibrofelt and other compressed vegetable fibre.

Some of these materials absorb moisture and need to be protected by means of waterproof paper or they must be made waterproof in the process of manufacture. This statement refers to hair felt, vegetable fibres and woods and to the mineral insulations like rock cork, waterproof lith, asbestos and others of similar nature. Waterproof lith is a mixture of flax fibres, 40 per cent by volume, and limestone rock made waterproof by means of petroleum vapor during manufacture. Sawdust has been used and shavings are still used in the cheaper constructions, in which case the material should be dried and protected from the absorption of moisture by means of waterproof building paper. Granulated cork can be used to advantage in special cases, but as a rule the preferred type of insulation is in a form of board of convenient size for erection on the wall, floors and ceilings. Waterproof lith is made in boards 18 in. × 48 in., and corkboard is usually 12 in. × 36 in. Thicknesses vary, but cork can be obtained from $\frac{1}{4}$ to 1 in. by eighths, and in $1\frac{1}{2}$, 2, 3, 4, and 6 in.

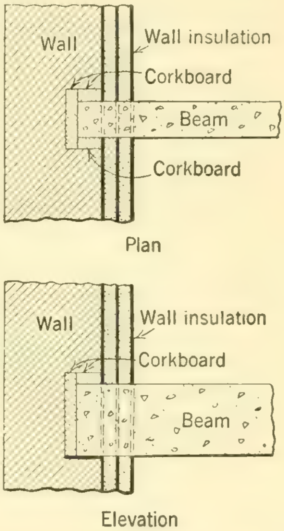
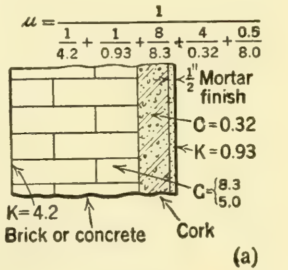


FIG. 134.—Continuous Wall Insulation.

Besides the consideration of waterproofness there are also the factors of fireproofness and protection from vermin. Sawdust has been used in ice-house construction for a long time, but it has nothing to recommend it except its cheapness, and shavings were used in the temporary cold storage buildings erected by the War Department in 1917 and 1918 (Fig. 287). In general, the kind of insulation should conform to the type of construction used in the building, and consideration should be taken of the probable life of the structure as well as the fractional part of the year that the building will probably be used for cold storage or other purposes where insulation is required. While cork is of vegetable origin it can be made waterproof by asphalt coating and to some extent

TABLE 45



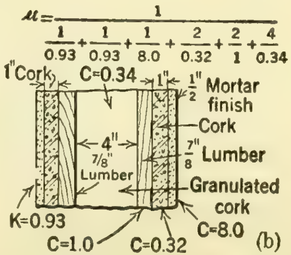


Figure	Thickness of Wall, Inches	Thickness of Cork, Inches	Brick Wall						Concrete Wall					
			Difference of temperature on the two sides of wall, degrees											
			1	10	20	40	60	100	1	10	20	40	60	100
(a)	8	2	0.1083	1.083	2.176	4.332	6.498	10.83	0.1164	1.164	2.328	4.656	6.984	11.64
		4	.0646	0.646	1.292	2.584	3.876	6.46	0.0673	0.673	1.346	2.692	4.038	6.73
		6	.0460	0.460	0.92	1.840	2.760	4.60	0.0474	0.474	0.948	1.896	2.844	4.74
		8	.0357	0.357	0.714	1.428	2.142	3.57	0.0366	0.366	0.732	1.464	2.196	3.66
	12	2	.0995	0.995	1.990	3.980	5.970	9.95	0.1101	1.101	2.202	4.404	6.606	11.01
		4	.0614	0.614	1.228	2.456	3.684	6.14	0.0653	0.653	1.306	2.612	3.918	6.53
		6	.0444	0.444	0.888	1.776	2.664	4.44	0.0464	0.464	0.928	1.856	2.784	4.64
		8	.0347	0.347	0.694	1.388	2.082	3.47	0.0359	0.359	0.718	1.436	2.154	3.59
	16	2	.0923	0.923	1.846	3.692	5.538	9.23	0.1047	1.047	2.094	4.188	6.282	10.47
		4	.0586	0.586	1.172	2.344	3.516	5.86	0.0633	0.633	1.266	2.532	3.798	6.33
		6	.0428	0.428	0.856	1.712	2.568	4.28	0.0453	0.453	0.906	1.812	2.718	4.53
		8	.0338	0.338	0.676	1.352	2.028	3.38	0.0353	0.353	0.706	1.412	2.118	3.53
(b)		2	.0449	0.449	0.898	1.796	2.694	4.49						
		3	.0394	0.394	0.788	1.576	2.364	3.94						
		4	.0351	0.351	0.702	1.404	2.106	3.51						
		6	.0287	0.287	0.574	1.148	1.722	2.87						

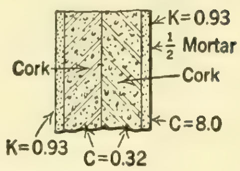
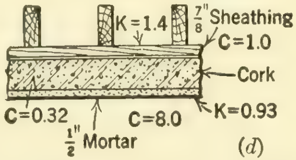
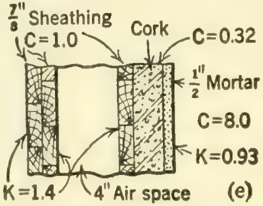
$\mu = \dfrac{1}{\dfrac{2}{0.93} + \dfrac{1}{8.0} + \dfrac{4}{0.32}}$  (c)	4	.0677	0.677	1.354	2.708	4.062	6.77					
	5	.0559	0.559	1.118	2.236	3.354	5.59					
	6	.0475	0.475	0.950	1.900	2.850	4.75					
	8	.0367	0.367	0.734	1.468	2.202	3.67					
$\mu = \dfrac{1}{\dfrac{1}{1.4} + \dfrac{1}{0.93} + \dfrac{1}{1} - \dfrac{0.5}{8.0} + \dfrac{2}{0.32}}$  (*d*)	2	.110	1.10	2.20	4.40	6.60	11.0					
	4	.0652	0.652	1.304	2.608	3.912	6.52					
	6	.0463	0.463	0.926	1.852	2.778	4.63					
 (e) $\mu = \dfrac{1}{\dfrac{1}{0.93} + \dfrac{3}{1.4} + \dfrac{3}{1} + \dfrac{0.5}{8.0} + \dfrac{2}{0.32}}$	2	.0798	0.798	1.596	3.192	4.788	7.98					
	4	.0533	0.533	1.066	2.132	3.198	5.33					
	6	.0400	0.400	0.800	1.600	2.400	4.00					

TABLE 46

SPECIFIC CONDUCTIVITY OF VARIOUS MATERIALS

Materials.	α_{32}	β	
Cast iron..........	330.00	−0.0004	Average values for gray iron. Variations with composition very great.
Wrought iron (unworked)........	450.00	−0.0009	See next below as indication of variation.
Wrought iron...... (worked)..........	240.00	−0.0006	
Steel (soft)........	300.00	−0.0003	See below as indication of variations.
Steel (mod. hard)...	240.00	−0.0003	
Steel (very hard)...	180	....	
Copper (pure)......	2400	−0.0002	Values given by different experimenters vary considerably. Probably due to variations in purity and condition.
Copper (commercial).....	2100.	+0.0001	
Brass (yellow).....	420	+0.0014	Varies greatly with composition.
Brass (red)........	540	+0.0008	
Aluminum (pure)...	750	+0.0003	
Aluminum with { 0.5% Fe, 0. 4% Cu }	1041	+0.0002	
Cylinder oil........	0.784	−0.0015	Naturally varies with kind of oil, cylinder oil not being a definite compound.
Water.............	2.6158	+0.0053	These values seem best authenticated. Authorities differ greatly.
Air...............	0.11989	+0.0017	Varies with humidity, etc.
Hydrogen.........	0.71933	....	
Fire brick.........	....	....	6.948 at 1300° F. Varies considerably with composition of brick.
"Insulating" materials........	0.4 to 1.2	....	Such materials as cork, cellular paper, asbestos mixtures, etc.

$$\alpha_t = \alpha_{32}\left\{1 + \beta\,(t-32)\right\}.$$

TABLE 47

VALUES OF $r_2 \log_e (r_2/r_1)$ FOR COVERINGS OF VARIOUS THICKNESSES ON DIFFERENT-SIZED PIPES

Pipe Size, Inches		Thickness of Covering: 1 in.		1½ in.		2 in.		2½ in.		3 in.		4 in.	
	r_1	r_2	$r_2 \log_e \frac{r_2}{r_1}$	r_2	$r_2 \log_e \frac{r_2}{r_1}$	r_2	$r_2 \log_e \frac{r_2}{r_1}$	r_2	$r_2 \log_e \frac{r_2}{r_1}$	r_2	$r_2 \log_e \frac{r_2}{r_1}$	r_2	$r_2 \log_e \frac{r_2}{r_1}$
½	0.420	1.420	1.730	1.920	2.920	2.420	4.235	2.920	5.660	3.420	7.165	4.420	10.400
¾	0.525	1.525	1.625	2.025	2.733	2.525	3.965	3.025	5.294	3.525	6.710	4.525	9.760
1	0.657	1.657	1.532	2.157	2.563	2.657	3.710	3.157	4.950	3.657	6.278	4.657	9.110
1¼	0.830	1.830	1.448	2.330	2.402	2.830	3.470	3.330	4.622	3.830	5.855	4.830	8.500
1½	0.950	1.950	1.400	2.450	2.322	2.950	3.340	3.450	4.445	3.950	5.620	4.950	8.170
2	1.187	2.187	1.335	2.687	2.194	3.187	3.145	3.687	4.177	4.187	5.262	5.187	7.658
2½	1.437	2.437	1.288	2.937	2.062	3.437	2.996	3.937	3.965	4.437	5.000	5.437	7.230
3	1.750	2.750	1.240	3.250	2.010	3.750	2.856	4.250	3.770	4.750	4.740	5.750	6.840
3½	2.000	3.000	1.216	3.500	1.959	4.000	2.770	4.500	3.647	5.000	4.580	6.000	6.590
4	2.250	3.250	1.190	3.750	1.913	4.250	2.700	4.750	3.598	5.250	4.450	6.250	6.385
4½	2.500	3.500	1.177	4.000	1.880	4.500	2.640	5.000	3.465	5.500	4.340	6.500	6.210
5	2.781	3.781	1.161	4.281	1.845	4.781	2.588	5.281	3.383	5.781	4.226	6.781	6.043
6	3.312	4.312	1.138	4.812	1.795	5.312	2.507	5.812	3.266	6.312	4.071	7.312	5.790
7	3.812	4.812	1.117	5.312	1.761	5.812	2.448	6.312	3.190	6.812	3.955	7.812	5.626
8	4.312	5.312	1.101	5.812	1.732	6.312	2.405	6.812	3.114	7.312	3.860	8.312	5.450
9	4.812	5.812	1.109	6.312	1.710	6.812	2.370	7.312	3.060	7.812	3.790	8.812	5.335
10	5.375	6.375	1.085	6.875	1.694	7.375	2.334	7.875	3.007	8.375	3.707	9.375	5.220
12	6.370	7.375	1.073	7.875	1.660	8.375	2.285	8.875	2.932	9.375	3.614	10.375	5.060
14	7.000	8.000	1.067	8.500	1.654	9.000	2.262	9.500	2.900	10.000	3.566	11.000	4.986
16	8.000	9.000	1.059	9.500	1.630	10.000	2.236	10.500	2.852	11.000	3.500	12.000	4.860
18	9.000	10.000	1.048	10.500	1.620	11.000	2.203	11.500	2.817	12.000	3.455	13.000	4.795

$$Q = \frac{t_1 - t_2}{\dfrac{r_s \log_e \frac{r_2}{r_1}}{C_1} + \dfrac{r_s \log_e \frac{r_3}{r_2}}{C_2} + \dfrac{r_s \log_e \frac{r_4}{r_3}}{C_3} + }$$ B.t.u. per square foot of outside area.

by the mortar finish. The mortar finish assists in making cork fireproof, but under any conditions it is a very slow burning material. At the present time corkboard is used in the majority of installations where flat surfaces are to be insulated. In the case of tanks, shell and tube brine coolers, etc., molded insulation can be used to advantage, but in many cases granulated cork can be packed around the body to be insulated and held in place by means of a container made of tongue and grooved lumber with waterproof paper between the layers. A good example of this construction is shown in Fig. 270 for ice tank construction.

TABLE 48

For Drinking Water Conditions

Heat leakage in B.t.u. per linear foot of pipe per deg. F. difference of temperature per hour

Size of Pipe	Union Lith	Hair Felt (Two Layers)	Cork-board	Size of Pipe	Union Lith	Hair Felt (Two Layers)	Cork-board
$\frac{1}{2}$	0.157	0.086	0.160	3	0.294	0.192	0.304
$\frac{3}{4}$	0.170	0.095	0.167	$3\frac{1}{2}$	0.355	0.211	0.330
1	0.180	0.107	0.178	4	0.367	0.228	0.345
$1\frac{1}{4}$	0.213	0.124	0.199	$4\frac{1}{2}$	0.376	0.247	0.385
$1\frac{1}{2}$	0.217	0.132	0.220	5	0.446	0.268	0.410
2	0.269	0.149	0.245	6	0.558	0.307	0.437
$2\frac{1}{2}$	0.272	0.167	0.291				

The use of corkboard for pipe line construction is not always the best practice. Where the pipe has a tendency to shake, the material should have the ability to give a little. Under these conditions hair felt gives satisfaction if properly applied, that is, if the material is kept from absorbing moisture as it will unless the outside waterproofing is erected in place correctly.

The Effect of Moisture.—The effect of moisture on the value of the coefficient of heat transfer is very well explained in the paper by L. F. Miller presented at the 14th Western meeting of the American Society of Refrigerating Engineers, May, 1927. In this paper it is shown by experiments that the increase in the value of k may be as much as 50 per cent. It is also mentioned in the paper, a fact that others have noticed, that the value of k varies with the actual temperature of the material subject to heat transmission. As a rule the refrigerating

engineer is not particularly interested in either of these factors. The insulation is waterproofed as far as possible and the engineer realizes that there is a tendency for moisture to work into the material where freezing will ultimately disintegrate it. The usual practice is to make the insulation waterproof, or as nearly so as is practical, and to use a factor in the calculations for heat leakage that will allow for irregularities in the erection and for the presence of moisture.

Erection.—Dead air spaces in the floor or the wall are of no value in cold temperature insulation, but are a source of considerable trouble due to condensate collecting in these spaces, as is found in hollow tile or the space between joists. Roof and floor slabs are preferred to tile, and where air spaces are found in the construction it is best to fill them with granulated cork or other insulating material. Care must be taken in the insulation of columns or tanks on the ground floor or freezing of the earth under the footings will result. Where cork is applied to a concrete ceiling it may be laid in the concrete forms, and a finish of portland cement mortar can be applied after the forms have been removed. The larger cold storage warehouses are now constructed with continuous outside wall insulation without break due to tying the floors to the walls (Fig. 144). In rooms where the humidity is high, and where there is a tendency for moisture to condense on the surface, the cork should be covered with some special preparation. The Armstrong Cork Co. uses an *asphalt mastic* finish $\frac{1}{8}$ in. thick, which is ironed on at the factory and is made tight at the joints on erection by means of a hot trowel. If nails are used in corkboard they should be galvanized, but some engineers prefer the wooden skewer as wood is a poorer conductor of heat. Pitch or tar should not be used in the erection of corkboard, but instead some odorless material which will not tend to taint the foodstuffs in storage. Other things being equal, the asphalt is better than the cement plaster because of the waterproofness of the asphalt. Moisture will penetrate any form of insulation if the conditions are favorable.

In erecting corkboard against brick or concrete walls the first process should be to prime the masonry thoroughly with at least one coat of some sort of emulsified asphalt primer. If such a priming coat is not applied to the walls, and the corkboard is erected against the walls in portland cement, the air which seeps through will carry moisture with it which may condense back of the corkboard and even enter the interstices between the granules of cork, thereby causing a disintegration of the cork.

The use of asphalt or cement plaster in cold storage construction is still not definitely settled. Cement gives greater strength than asphalt

does, but the latter is waterproof and costs considerably less than cement mortar. Cement plaster will crack, therefore the plaster finish should be applied in two coats and score marks should be made every four feet.

Protecting and Finishing Cork Insulation.—A finish for cork insulation, in order to have a universal application to all conditions under which such insulation is used, should have the following properties:

1. It should bond securely to the face of the cork under all conditions of service and use.

2. It should remain intact—that is, should not develop cracks regardless of the temperature changes to which it is subjected. In other words, the continuity of the finish should not be broken.

3. It should be both air-proof and moisture-proof in order to insure the maximum insulation value of the cork sheet.

4. It should be sanitary—not absorbent of odors—and it should have a finish that may be cleaned readily by steam or hot water without damage.

5. It should be odorless itself, and in setting it should give off no fumes or noxious odors, as it must be used frequently in confined spaces.

6. It should be of such a character as to take decoration, as a paint or an enamel, should a paint finish be desired.

7. Its application should be within the range of the abilities of one of the usual building trades without special equipment or special knowledge.

8. It must be economical.

For the great majority of work there has been no alternative in the selection of the finish. The choice has been limited to the ordinary plaster, not because it was satisfactory and fully met service conditions, but because it was the only material available.

Plaster, however, is objectionable, and has proved unsatisfactory in many particulars. In the first place it does not bond well with cork. Cork is without suction or capillarity, and a surface that lacks this property is difficult to plaster on with any assurance of good adhesion. There are many instances where plaster has fallen off the cork sheets, and has had to be replaced with the addition of wire mesh or expanded metal in order to hold it in place.

Plaster not only develops cracks, but the very conditions of its use make the development of cracks inevitable. Cork sheets are almost always set up in hot asphalt, which is used as a cementing medium, although asphalt is not a a solid but is in reality a fluid. When the wall against which the cork is laid expands or contracts, some slight movement of the cork takes place. When this occurs the rigid plaster on the

face of the cork generally yields by cracking. It is claimed also that plaster is neither air-proof, moisture-proof nor sanitary since it will absorb both moisture and odors.

Standard Construction.—The following specifications and construction[4] represent good practice for pipes, fittings, tanks and building construction.

Insulation of Refrigerated Piping.—After all the pipe lines have been tested and have been made tight, they should be covered with cork covering and fitting covers of the proper thickness, as outlined in the following schedules:

For temperatures from 0 deg. F. to 25 deg. F.	Brine thickness.
For temperatures from 0 deg. F. to − 35 deg. F.	Special thick brine.
For temperatures from 25 deg. F. to 45 deg. F.	Ice water thickness.

All foreign matter, such as plaster, should be removed from all pipe surfaces before covering is applied. Under no circumstance shall pipes be covered while wet or in a frosted condition.

Sectional covering shall be used on all pipes 8 in. and smaller for brine thicknesses, 6 in. and smaller for special thick brine, and 10 in. and smaller for ice water thickness.

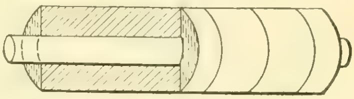


FIG. 135.—Pipe Insulation.

Sectional covering shall be applied with all end joints broken by making one-half of the first section 18 in. long and leaving the other half 36 in. in length (Fig 135). All longitudinal joints shall be on the top and the bottom of the pipe and not along the sides. All covering shall be applied with waterproof cement on all joints and wired in place with copper-clad steel wire, using not less than six wires to a section.

For brine thickness molded fitting covers shall be used on all screwed and flanged fittings 6 in. and smaller. In the case of special thick brine, molded fitting covers shall be used on standard screwed fittings 5 in. and smaller, ammonia and extra heavy screwed fittings $4\frac{1}{2}$ in. and smaller, and all flanged fittings 4 in. and smaller. For ice water thickness molded fitting covers shall be used on standard and extra heavy screwed fittings 6 in. and smaller, standard flanged fittings 4 in. and smaller.

Molded fitting covers shall be applied with waterproof cement on all joints and wired in place with copper-clad steel wire, using not less than four wires on screwed fittings and six wires in flanged fittings

[4] The following covers the Armstrong Cork and Insulating Co. practice.

(Figs. 136 and 137). All spaces between covers and fittings shall be filled carefully with brine putty.

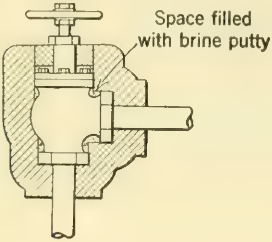


FIG. 136.—Insulation for an Angle Valve.

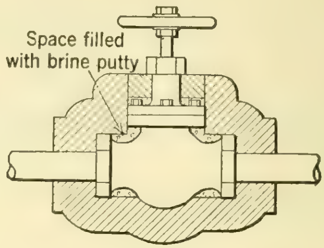


FIG. 137.—Insulation for a Globe Valve.

After the covering has been applied, all seams and chipped edges shall be filled with a suitable seam filler so as to leave a smooth and even surface. The whole covering shall then be given one good coat of cork covering paint.

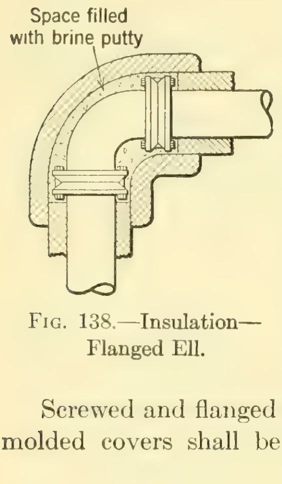


FIG. 138.—Insulation—Flanged Ell.

All pipes larger than the sizes listed for sectional covering shall be covered with cork lagging weighing not less than 1.2 lb. per foot, board measure, and beveled to the proper diameter. All joints shall be broken by starting with alternate 18 ins. and 36 in. lags. Waterproof cement shall be used on all joints, and the lags shall be wired in place with copper-clad steel wire using not less than six wires to every three feet of lagging.

Screwed and flanged fittings larger than the sizes just mentioned for molded covers shall be covered with fitting covers made from cork

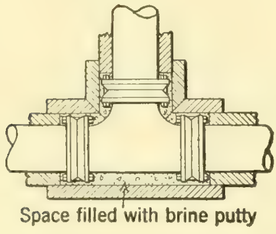


FIG. 139.—Insulation—Flanged Tee.

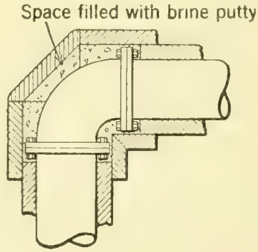


FIG. 140.—Insulation Flanged Ell of Large Size, with Cover.

lagging weighing not less than 1.2 lb. per foot, board measure, and beveled to the proper diameter. All lagging shall be assembled with waterproof cement on all joints and wired in place with copper-clad steel wire, using not less than six wires per fitting (Figs. 138 and 140).

Cylindrical Brine Coolers and Tanks.—The following table gives the thickness of cork lagging that has been found economical for the temperatures noted:

	Inches
Below 5 deg. F.	6
5 deg. to 20 deg. F.	5
20 deg. to 32 deg. F.	4
32 deg. to 55 deg. F.	3
55 deg. to 65 deg. F.	2

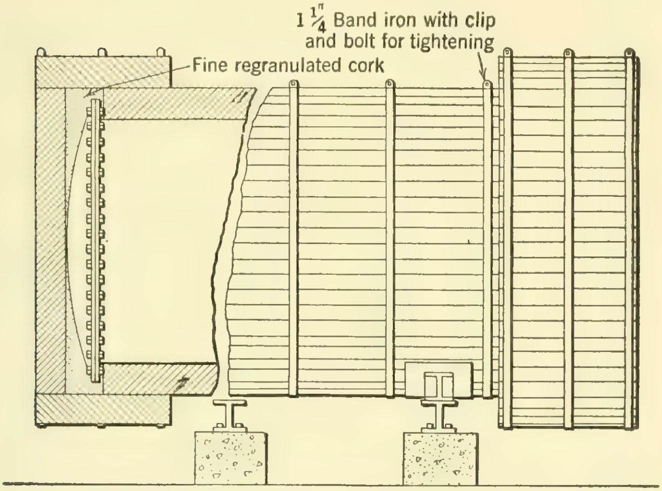


FIG. 141.—Insulation—Horizontal Brine Cooler.

The cylindrical surface of the tank will be insulated with one layer of cork lagging of the proper thickness. The lagging shall be beveled to the proper diameter and shall have a mineral rubber finish ironed on at the factory to both the inner and the outer surfaces. The lags shall be applied with waterproof cement on all joints and secured with $1\frac{1}{4}$-in. bands of 16-gage galvanized iron, drawn up tight by means of bolts through clips riveted to the ends of the bands. The bands shall be spaced not more than 12 in. apart. The lags shall be applied with broken transverse joints. All spacing between the tank and the lagging shall be filled with brine putty.

The flanged ends of the cooler or the tank shall be insulated with a suitable disk of cork lagging supported by a layer of cork lagging over

the flanges. This flange lagging shall project beyond the head of the tank a distance equal to the thickness of the disk and shall have a bearing on the body lagging of not less than the overhand required to enclose the disk, tank head and flange bolts. The flange lagging shall be applied with waterproof cement on all joints and secured by not less than three galvanized iron bands as described.

Walls.—It is recommended that the walls of ice storage rooms be built of hard pressed brick. The roof must be left around all walls for a

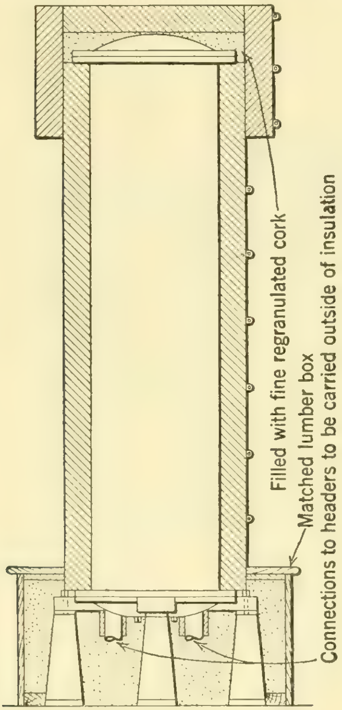


FIG. 142.—Insulation—Vertical Brine Cooler.

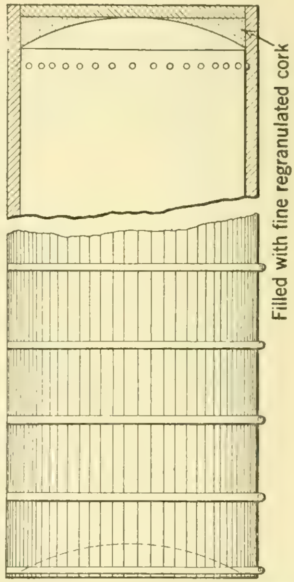


FIG. 143.—Insulation—Riveted Brine Cooler.

space equal to the thickness of the wall insulation plus 1 in., so that the wall insulation may be continuous to its connection with the roof insulation, as shown in Fig. 145. Any spaces between the insulation and the edge of the roof slab shall be filled with cement grout.

The wall surfaces must be smooth and true, and thoroughly cleaned of all dirt, dust, plaster, loose mortar, etc. If rough and uneven, the wall surfaces must be plastered with portland cement plaster, mixed one part of cement to three parts of clean sand, applied thick enough to

true up the wall surface and not less than $\frac{1}{4}$ in. thick at the high points. The surfaces of the walls (if plastered, after the plaster has dried thoroughly) must then be painted with a good coat of asphalt paint.

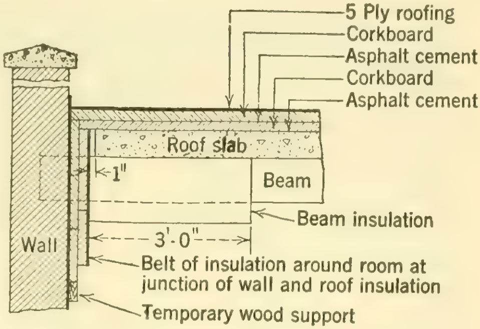


FIG. 144.—Insulation—Wall and Floor.

Prepare a suitable amount of asphalt cement. Use plain unscored corkboard. Dip each piece—one side, the end that is to butt the board that is already in place, and the lower edge—in asphalt, then put it in place quickly. The first row of corkboard must be erected on the wall at the floor on a level line, and should be carried clear around the room if all four walls are to be insulated. The boards must be kept in perfect alignment, in order that the joints in the upper rows may fit close and tight. As each board is put in place care must be taken to butt it tightly against the adjoining boards. All vertical joints must be broken, or must be staggered, as shown in Fig. 145.

A second course shall be laid as follows: Saw some of the boards lengthwise down the center, so as to have enough pieces of half the regular width to make one row around the room. Start at the floor line with these narrow pieces, staggering joints with the floor insulation as shown in Fig. 144. Dip each piece—the plain side, the end that is to abut the board that is already laid in place, and the lower edge—in asphalt, and then put it in place quickly, drawing it up tightly against the first course of corkboard with either galvanized iron nails or wooden skewers. Care must be taken to prevent the vertical joints of the second course from coming directly over the vertical joints in the first course. Then continue the erection of the second

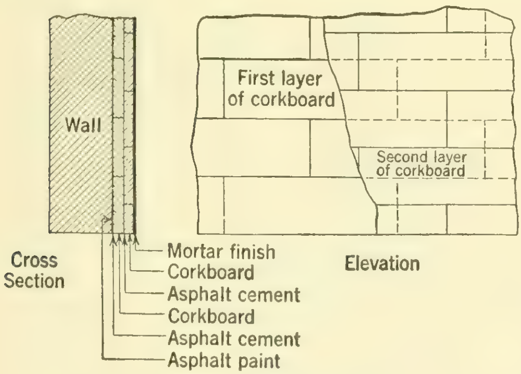


FIG. 145.—Insulation—Wall.

layer of corkboard, using sheets of full length and width. Care must be taken to keep the first row in perfect alignment so that the upper rows may be close and tight, as the efficiency of the insulation depends upon getting tight joints. At the floor and roof and in the corners of the room the two layers of corkboard should be joined in the manner shown in Fig. 146.

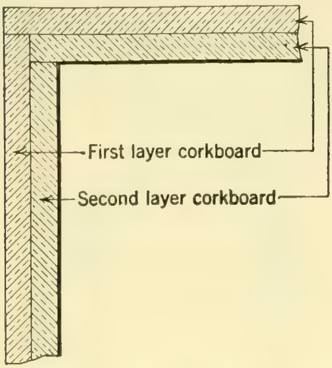


FIG. 146.—Insulation—Joint at Wall and Ceiling.

The nails or skewers should be driven in obliquely, two to the square foot. They must be long enough to reach well into the first course, but not through the wall. Two nails or skewers, one near each end, should be driven part way into each board before it is put in place and used to hold the board when dipping it in the asphalt cement.

All joints and nail heads shall be filled with a plastic seam filler. Care must be taken to see that the joints between the wall and the roof insulation are staggered and made perfectly tight by sealing with hot asphalt in the manner shown in Fig. 146. In case the roof insulation is to be laid before the walls are insulated, a belt of insulation must first be erected around the top of the walls in order to insure the proper connection of wall and roof insulation. This belt should be wide enough to extend slightly below the roof beams and should rest on a wood support temporarily secured to the wall, as shown in Fig. 146, until the wall insulation is put up.

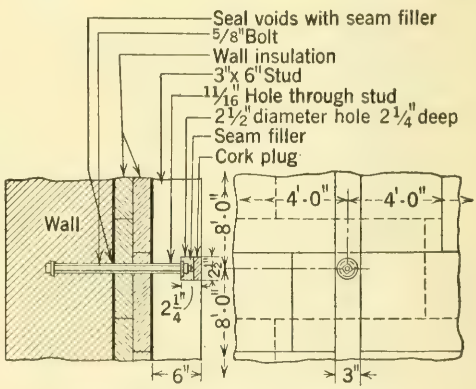


FIG. 147.—Insulation—Buckstay.

Where the height of the room is over 14 ft., *buckstays*, or studding, 3 × 6 in., must be erected on 4-ft. centers horizontally and 8-ft centers vertically to a depth of 8 in. in the wall and extending through the insulation and approximately 4 in. beyond. Any voids around the bolts at the face of the brick wall shall be

packed with seam filler before the corkboard is erected. The buckstays will fasten securely 3 × 6 studs erected on edge and drawn up tight against the insulation, as shown in Fig. 147.

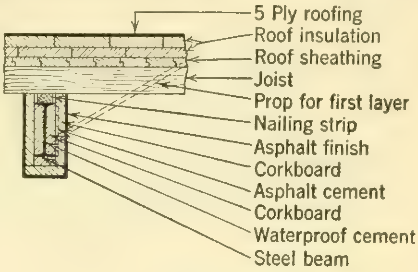


FIG. 148.—Insulation—Steel Beams.

Beams.—The *steel beams* supporting the roof must be insulated for a distance of 3 ft. out from the wall line with the same thickness of insulation used on the walls. The first layer is to be erected against the beam with waterproof cement and propped in place until the cement sets, as shown in Fig. 148. *Concrete beams* are to be insulated in the same manner as the wall and for the same distance as specified for steel beams.

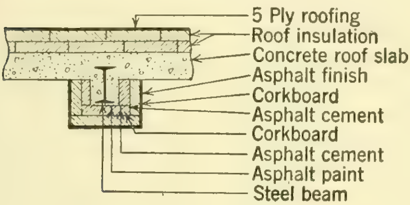


FIG. 149.—Insulation—Concrete Beams.

Roof.—The concrete or wood roof, upon which the corkboard is to be applied, must first be cleaned. All dirt, plaster, dust, loose mortar, whitewash, etc., must be removed carefully. In the case of a concrete slab, if the surface is not reasonably smooth, it should be covered with a light coating of portland cement in order to make an even surface against which to apply the corkboard. Prepare a suitable amount of asphalt on the basis of approximately two and one-quarter pounds for each square foot of roof surface.

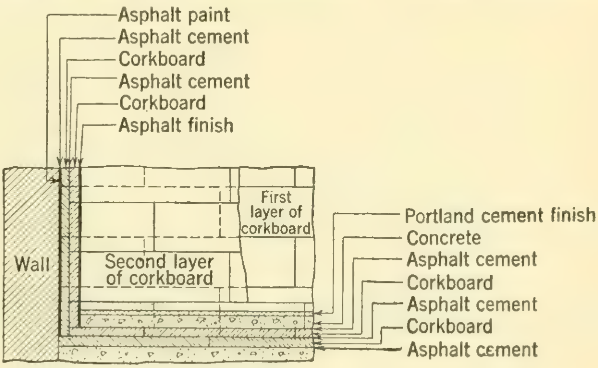


FIG. 150.—Insulation—Wall and Floor.

Use plain corkboard; that is, corkboard that has not been scored. Pour out on the roof only sufficient asphalt to lay one board at a time. As each sheet is put down push it along a few inches in the melted asphalt so that sufficient asphalt

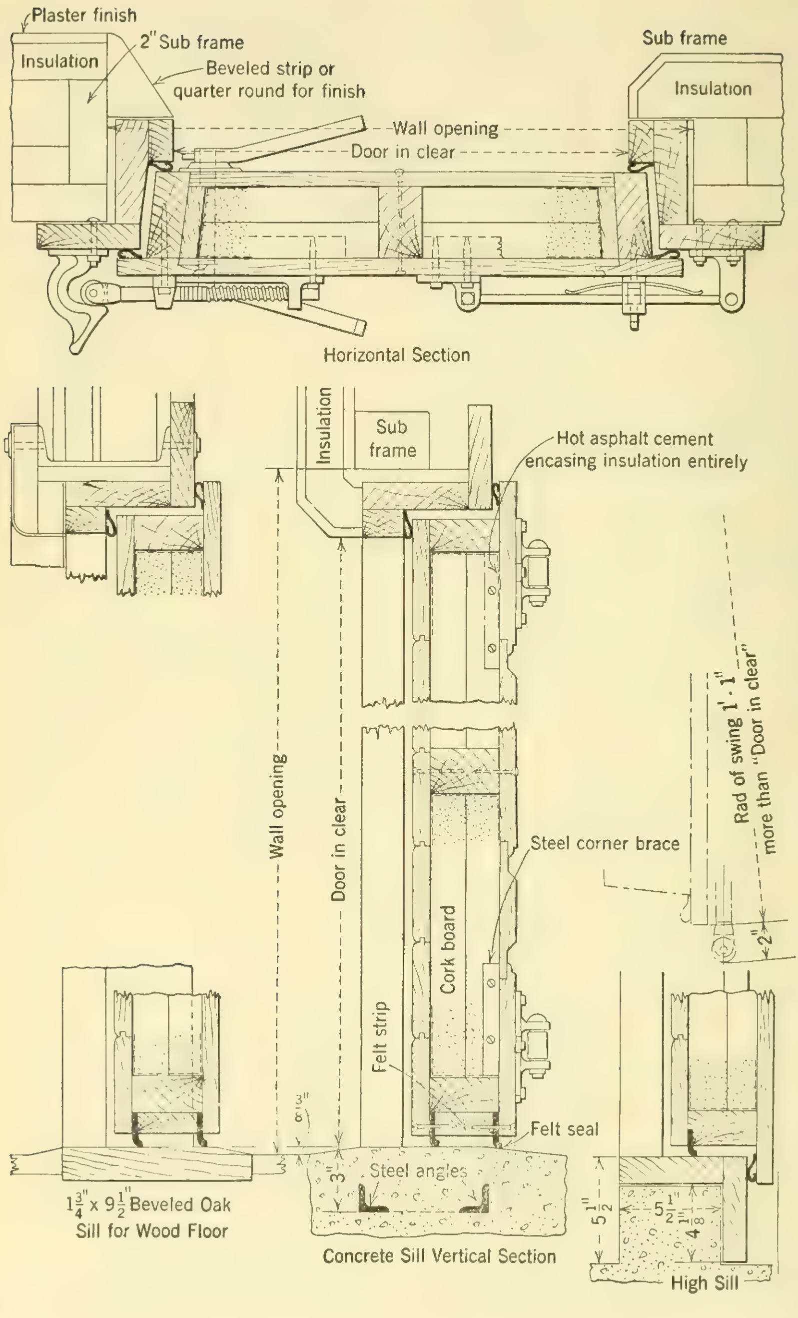


FIG. 151.—Insulation—Cold Storage Door.

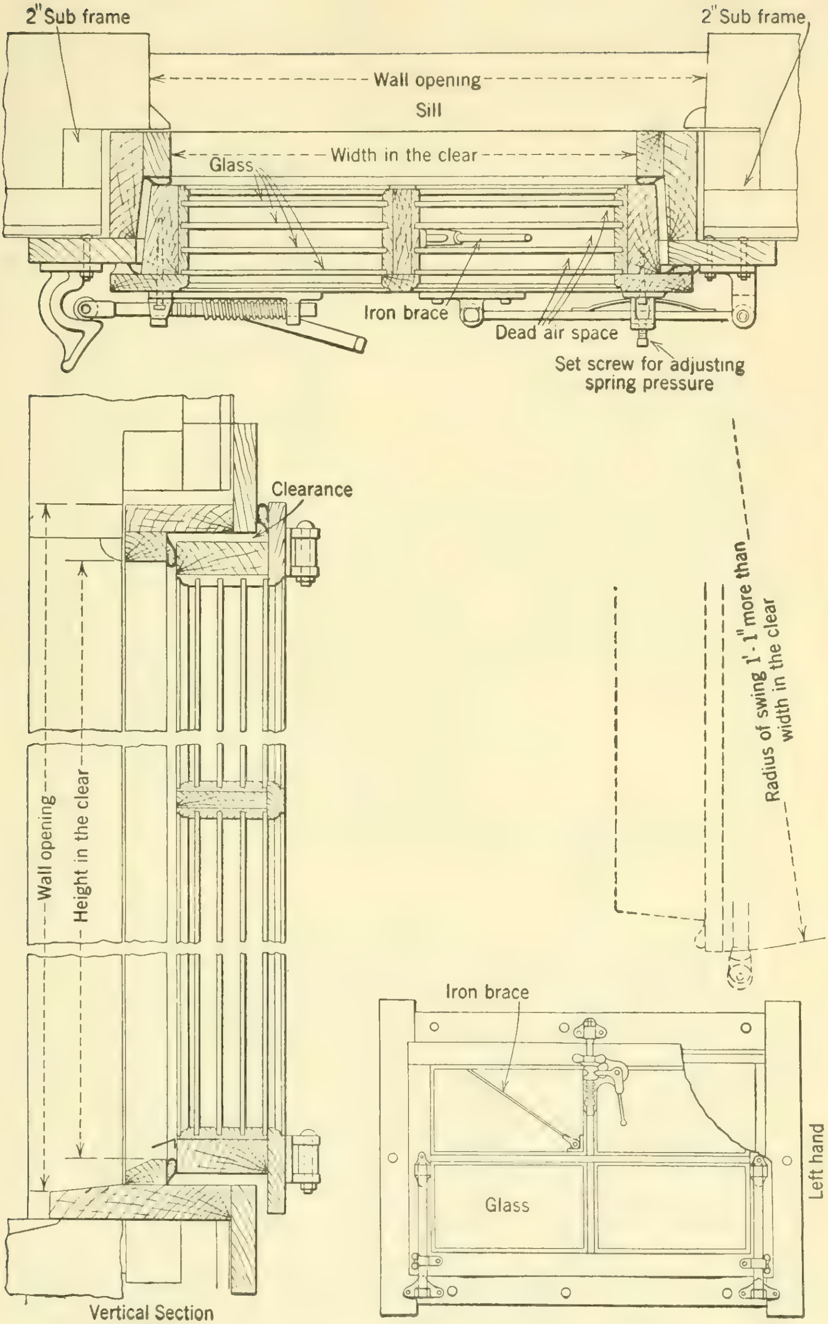


FIG. 152.—Insulation—Cold Storage Window.

will accumulate on the edge of the board to seal the joints effectively. Put down the first row along one side of the roof. Be careful to stagger

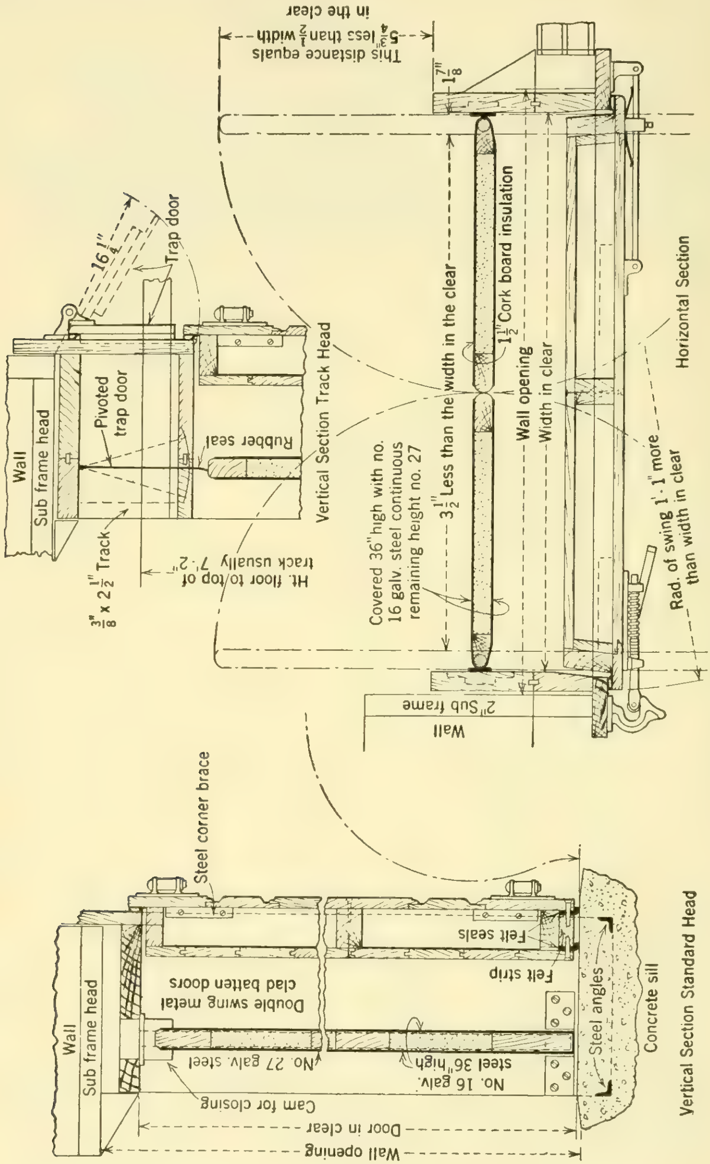


Fig. 153.—Insulation—Cold Storage Door.

joints with the wall insulation and to seal these contacts with hot asphalt. Take care to keep the boards in perfect alignment, so as to

secure tight fitting joints in the rows to follow. Break the joints in the second row and the second course in a manner similar to that used in wall construction, but be sure that the short joints of the second course do not come over the short joints of the first course. Especial care must be taken to insure that the joint between the roof and the wall insulation is perfectly tight and well sealed with asphalt.

Floor.—The corkboard shall be applied in a manner similar to that used in roof and wall insulation, after which the surface shall be flooded with hot asphalt to a thickness of at least $\frac{1}{8}$ in. by simply pouring the

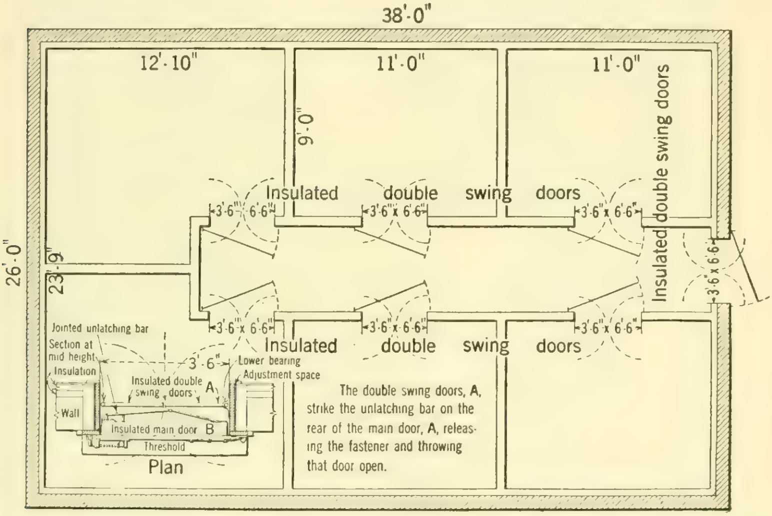


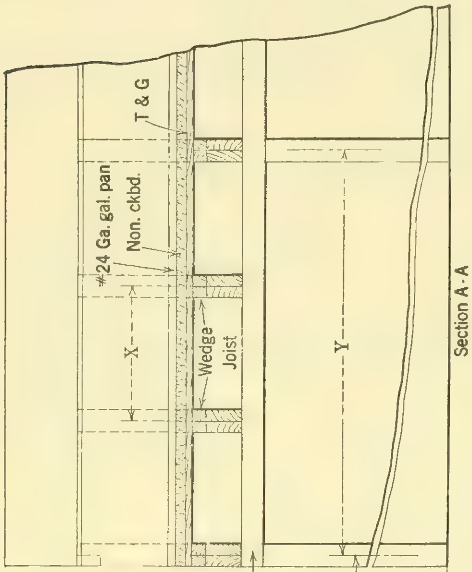
FIG. 154.—Cold Storage Doors—The Double Swing Door.

molten asphalt uniformly over the surface. On this surface a 4-in. concrete wearing floor shall be laid down.

Tables 50 and 51 give dimensions and shipping weights for corkboard. The design and construction of cold storage doors are shown in Figs. 151, 153 and 154, and a typical cold storage window by Fig. 152. Bunker room construction, showing the opening for the warm and the cold air, and the insulation required are shown in Figs. 155 and 156.

The Economic Thickness of Insulation.—It has been the custom of engineers to use the rough rule of 1 in. of corkboard for every 10 deg. F. difference in temperature between that of the cold storage room or warehouse and the mean temperature of the atmosphere. This usually resulted in thicknesses of insulation of from 3 to 4 in. in the case of cooler walls and 6 to 9 in. for freezer rooms, depending on the extreme limits to be experienced in the work. Like most rules of thumb, the thickness

to be used has no reference to the cost of the power consumed or of the investment either in the building or in the equipment directly con-



Note.- This table is actual-add 25% for waste (lumber)

	Width of Room	Posts Size	Posts No	Posts ℄	Joists Size	Joists No.	Joists ℄	Joists ℄	Plates Size	Plates No	Plates L	Wedge Size	Wedge No	Wedge L	Studs Size	Studs No	Studs L	T & G 7/8 T-G	Bunker Curb	Baffle Cap	Pan #24 Ga	Corkboard 1" or 2" Baffle Wall	Corkboard 2" Non. Bunk Floor
Coil only	6	4x4	To be calculated for each room	6	2x4	5	24	6	4x4	2	Room length	½(2x4)	5	3'-4"	2x4	5	1'10"	5 sq. ft	In all cases 2"x6" each length of room	1"x2" or 1"x3" depending upon the ckbd. in baffle	5 sq ft	1 0 sq ft	3 5 sq. ft
	8	4x4		6	2x6	5	24	8	4x4	2		½(2x6)	5	4'-10"	2x4	.5	1'-8"	6.5			6.5	1 3	4 9
	10	4x4		6	2x6	5	24	10	4x4	2		½(2x6)	5	6'-1"	2x4	.5	1' 8"	7 75			7 75	1 3	6 2
Meat & Coil	6	4x6		6	2x6	5	24	6	4x6	2		½(2x6)	5	3'-4"	2x4	5	1'-8"	5			5	1 3	3 5
	8	4x6		6	3x6	5	24	8	4x6	2		½(3x6)	5	4'-10'	2x4	.5	1'-8"	6 5			6 5	1.3	4 9
	10	4x6		4	4x6	5	24	10	4x6	2		½(4x6)	5	6' 1"	2x4	5	1'-8"	7 75			7 75	1.3	6.2
				Y			X											★			★	★	

Width of Room	Air Ducts Warm 18% W	Air Ducts Cold 15%W in clear	Weight to be carried Coil only	Weight to be carried Coil & meat	W. of Bunk.
[illegible]	1'-1"	1'-1" (horizontal)	222	592	3'-10"
8	1'-5"	1'-5" (horizontal)	310	825	5'-2"
10	1'-9"	1'-10" (horizontal)	386	1029	6'-5"

Notes:

1-"L" Length of post to be calculated for each room.

2-Plates, joists & wedges are shown built up from 2" & 3" material. Solid may be used if convenient.

3-The amounts shown in columns marked ★ corresponds to the stud height shown opposite. Any change in this height will necessitate a a corresponding change in the columns mentioned above.

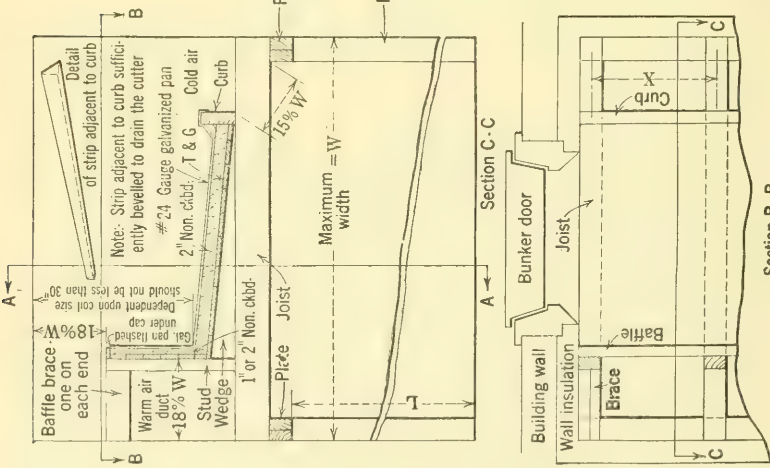
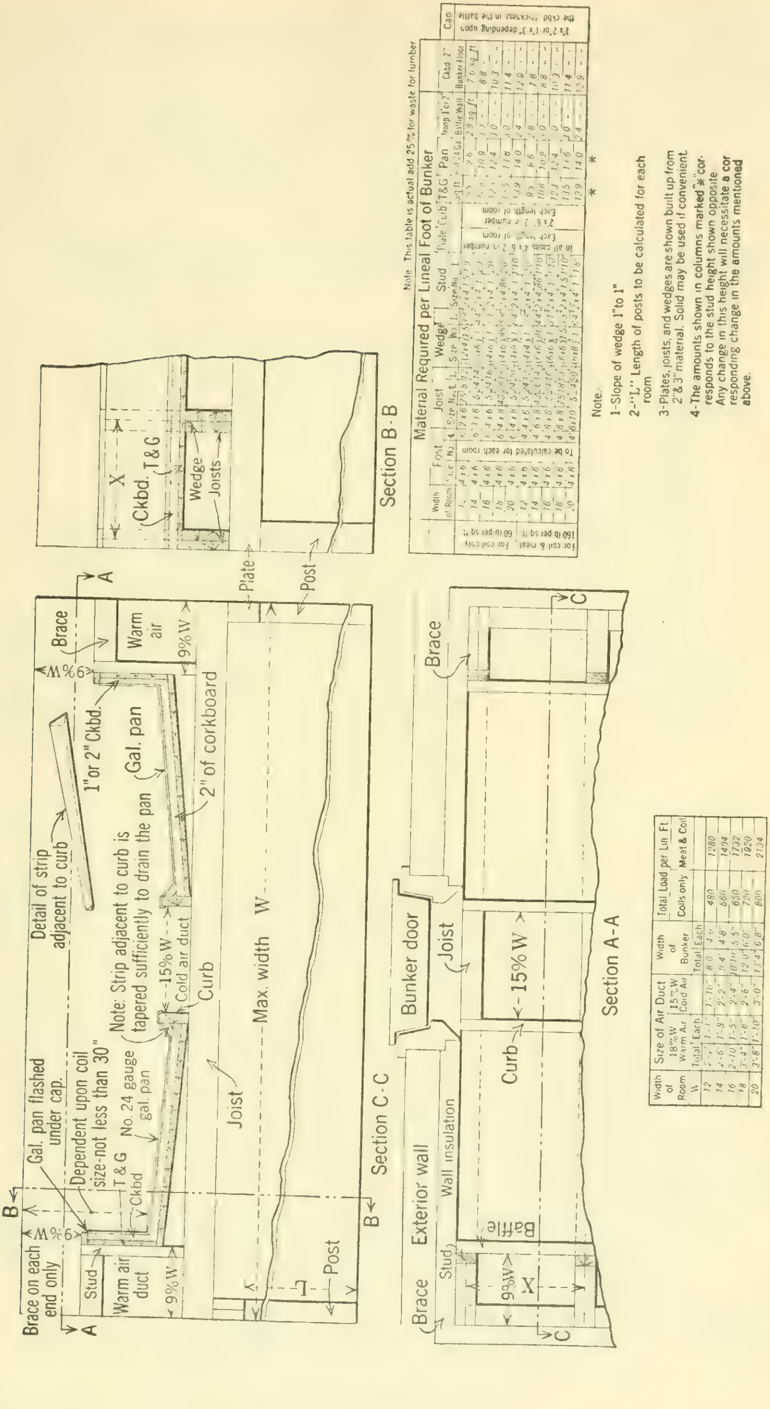


FIG. 155.—Insulation—Single Bunker.

cerned in the production of refrigeration. It is evident, when one considers the subject, that if a ton of refrigeration costs when delivered to



Width of Room W	Size of Air Duct 18%W Warm Air Total	Each	15%W Cold Air	Width of Bunker Total	Each	Total Load per Lin Ft Coils only	Meat & Coil
12	[illegible]	1'-1"	1'-10"	8'0"	4'0"	480	1280
14	2'-6"	1'-3"	2'-2"	9'4"	4'8"	660	1494
16	2'-10"	1'-5"	2'-4"	10'10"	5'5"	650	1732
18	3'-4"	1'-8"	2'-6"	12'0"	6'0"	720	1920
20	3'-8"	1'-10"	3'-0"	13'4"	6'8"	800	2134

FIG. 156.—Insulation—Double Bunker.

the room where it is to be used—including all operating, interest and depreciating charges—only \$1 per 24 hours, then the thickness of insulation which it is economical to use in order to reduce the cost of refrigeration must be smaller than in the case where the charge of \$2 or \$3 would be made. The cost of power, of water, and of labor are decided factors that must be considered if the thickness of insulation is to be chosen

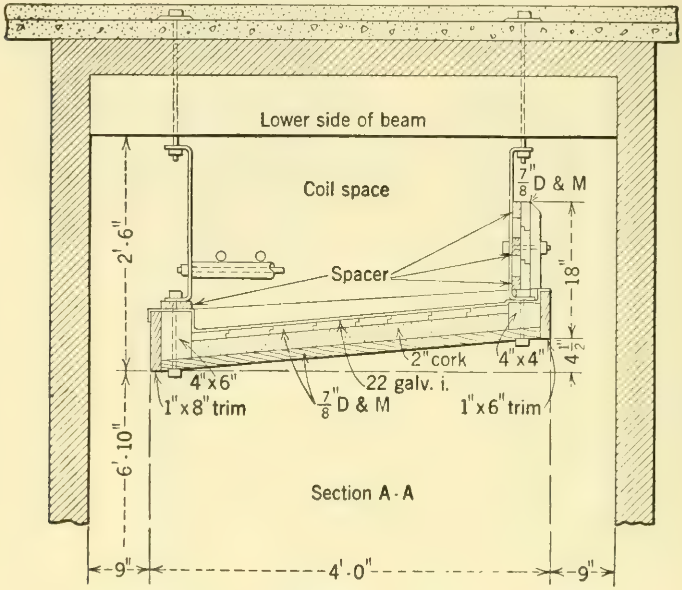


FIG. 157.—Insulation—Bunker Suspended from Ceiling.

which will give the greatest return on the investment. The factors influencing a choice of the economic thickness include then:

1. The cost of a ton of refrigeration delivered, per 24 hours.
2. The cost of the insulation applied, per year.
3. The cost of the refrigerating equipment not included in item 1, and particularly the room piping, etc., per year.
4. The value of the space occupied by the insulation, per year. Considering the case of a wall, ceiling or floor,[5] these losses may be expressed in the following manner:

Let x = the thickness of the insulation, in inches;
C = the coefficient of conduction of the insulation, per 1 in. thick, per hour;

[5] P. Nicholls, The Economic Thickness of Insulation in the Refrigerating Field, Refrigerating Engineering, 1923.

F = the yearly load factor;
B = the cost in dollars for the insulation applied, per 1 in. thick, per 1 sq. ft. of surface;
A = the cost in dollars per ton of refrigeration per 24 hrs., delivered;
G = the cost in dollars per ton of refrigeration of the machinery, etc., not included in A;
I = the interest rate as a per cent on the insulation investment;
R = the repair cost per year as a per cent of the insulation first cost;
Y = the life of the insulation, in years;
I', R', Y' = Similar values applied to machinery, etc., in G;
t_a = the temperature of the outside air, deg. F., as an average for the period of operations;
t_m = the maximum temperature of the outside air, in deg. F.;
t = the temperature of the cold storage room, in deg. F.;
t_p = the temperature of the refrigerant in the piping, in deg. F.;
S = the value per year of 1 cu. ft. of space in the cold storage room;
u = the coefficient of heat transfer of the wall, per hour, for the materials of construction other than the insulation as given by the usual formula:

$$\frac{1}{u} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{T_1}{C_1} + \frac{T_2}{C_2}, \quad \text{etc.}$$

Using these symbols, the separate costs will be:

1. The cost per year of the heat leakage through the insulation

$$= \frac{t_a - t}{\frac{1}{u} + \frac{x}{C}} \times \frac{24 \times 365 \times F \times A}{288{,}000}$$

2. The cost of the insulation per year, per square foot per 1 in. thick

$$= B \times x\left(\frac{I}{100} + \frac{R}{100} + \frac{1}{Y}\right)$$

3. The cost per year of the investment required to offset the heat leakage through the insulation

$$= \frac{t_m - t}{\frac{1}{u} + \frac{x}{C}} \times \frac{24}{288{,}000} \times G\left(\frac{I'}{100} + \frac{R'}{100} + \frac{1}{Y'}\right)$$

4. The cost of the space occupied by the insulation per year

$$= \frac{Sx}{12}$$

As a rule the cost of insulation, applied, can be expressed by the formula

$$B = \frac{C'}{x} + B' \text{ dollars per board foot,}$$

where

C' is the cost of finish, plaster, nails, labor and overhead per square foot;

and

B' is the cost of the insulation delivered to the job.

Also, if

P is the cost in dollars per square foot of refrigerating piping installed in the cold room as the equipment represented by G,

then

$$G = \frac{12{,}000 \times P}{k \times (t - t_p)},$$

where

k = the coefficient of heat transfer for the piping per *hour*.

Then if Z = total cost per year,

$$Z = \frac{t_a - t}{\frac{1}{u} + \frac{x}{C}} \times \frac{365FA}{12{,}000} + \left(\frac{C'}{x} + B'\right) \times \left(\frac{I}{100} + \frac{R}{100} + \frac{1}{Y}\right)$$
$$+ \frac{Sx}{12} + \frac{t_m - t}{\frac{1}{u} + \frac{x}{C}} \times \frac{1}{12{,}000} \times \frac{12{,}000 \times P}{k \times (t - t_p)} \times \left(\frac{I'}{100} + \frac{R'}{100} + \frac{1}{Y'}\right).$$

For a minimum, $dZ/dx = 0$, so, by differentiating and putting this equal to zero and solving for x, the result becomes,

$$x = 1.74\sqrt{\frac{A(t_a - t)F + \frac{0.327P}{k(t - t_p)}\left(I' + R' + \frac{100}{Y'}\right)(t_m - t)}{B'\left(I + R + \frac{100}{Y}\right) + 8.3S} \times C} - \frac{C}{u}$$

As an example of the manner in which the value of x may be calculated, the following values may be taken:

The wall may be considered as the equivalent of 12 in. of brick with a $\frac{1}{2}$-in. mortar finish on the cork.

Let I = 6 per cent;
I' = 6 per cent;
R = 3 per cent;

$$R' = 3 \text{ per cent};$$
$$Y = 15 \text{ years};$$
$$Y' = 8 \text{ years};$$
$$k = 1.5 \text{ B.t.u. per hour};$$
$$C = 0.35 \text{ B.t.u. per hour};$$
$$t_{average} = 50 \text{ deg. F.};$$
$$t = 0 \text{ deg. F.};$$
$$t_{max} = 95 \text{ deg. F};$$
$$(t - t_p) = 10 \text{ deg. F.};$$
$$F = 1.0;$$
$$A = \$2.00;$$
$$S = \$0.40 \text{ per cu. ft.};$$
$$t_{max} - t = 95 \text{ deg. F.};$$
$P = \$4.00$, the cost of piping per square foot installed, plus all accessories.

$$B = \left(\frac{0.72 + 0.065x}{x}\right);$$

$$G = \frac{288{,}000 \times P}{24 \times k \times (t - t_p)}$$

Placing these values in the equation the value of x becomes 7.0 in. Using a value of \$0.10 for S the value of x is changed to 11.5 in. Solving the general equation for the cost with varying values of x, the curve in Fig. 159 is obtained.

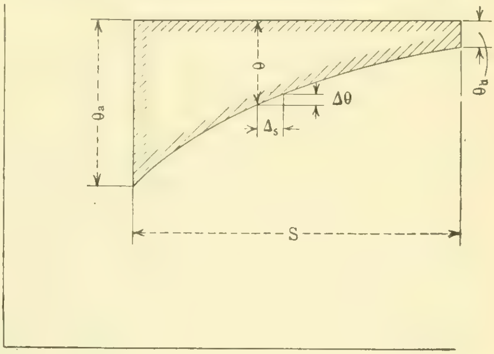


FIG. 158.—Hausbrandt's Formula.

Heat Transfer; Apparatus.—More important than the values of heat transfer of building materials are those pertaining to piping, condensers, and coolers of various sorts. The correct design of every portion of the refrigerating plant depends on the proper use of the various heat transfer coefficients. Unfortunately, these values are frequently little better than a guess, in which case the selected values are so liberal as to enable the load imposed on the apparatus to be carried under any set of circumstances, which are very varied in commercial work.

For piping and built-up heat transfer apparatus the total heat

passing through the surface is given by the well-known formula:

$$Q = Ak(t_2 - t_1)$$

where

A = the surface area in square feet;
k = coefficient of heat transfer in B.t.u. per degree difference in temperature per hour per square foot;
Q = B.t.u. per hour;
$(t_2 - t_1)$ = the temperature difference on the two sides of the surface.

Until recently it has been understood by refrigerating engineers that the area of piping is to be taken as the outside area, but the recent test code of the American Society of Mechanical Engineers gives the area as figured from the side of the refrigerant. However, this value as figured is a fictitious value in many cases. The flooded atmospheric and the flooded double-pipe condenser (for example) with at least the lower half of each pipe filled with liquid has its *effective condensing* area reduced in proportion.[6] The double-pipe and the atmospheric condenser with the hot gas entering at the top will certainly lose in effective liquefaction area as con-

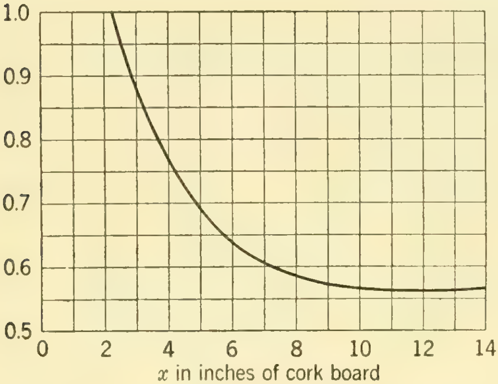


FIG. 159.—Problem—Economic Thickness of Insulation.

[6] The feeling prevails among refrigerating engineers that heat transfer is greater from the metal walls of the condenser to the condensate and then to the gas than from the walls to the saturated refrigerant. In the ammonia condenser the area exposed to the superheated gas is comparatively small in amount. In the bleeder type, for example, this amount is frequently limited to two pipes and only under exceptional circumstances does it increase to three pipes. The remainder of the condenser is in a wet condition with beads of liquid condensate forming on the upper side and running down the inside or dropping to the bottom of the pipe. The refrigerant is *not* exposed to a dry surface but one constantly wet with the liquid. Except in that condenser design which forces the liquid through the pipes at a high velocity—thereby continually breaking down the surface film—there is no advantage, and in fact there is a *positive disadvantage* in permitting the liquid to remain on the surface of the condenser. Once condensed the refrigerant should be drained off as quickly as conditions will permit. In this respect the theoretical design of ammonia condensers follows closely that of the better known and better designed steam surface condensers.

densation occurs and the liquid condensate flows from pipe to pipe into the liquid receiver. The loss of effective surface (loss in the sense of decreased heat transfer due to the layer of liquid ammonia) might be easily 50 per cent or more of the surface in the lower tubes. This trouble is even greater in the case of the drip type condenser, which has a liquid flow in many of the pipes counter to the flow of the compressed gas discharge from the compressor, and always has most of the work of the condenser performed in the upper pipe when cold condensing water is used or in the upper two or three pipes when large amounts of warm water are used.

Another disturbing factor in condensers is the presence of air. In Orrok's[7] tests on steam condensers it was found that the presence of air reduced the value of k to about $\frac{1}{5}$ of the value obtained with an air-free surface.[8] However, all commercial condensers have air present to a greater or a lesser degree, although there is more care now (1927) than

[7] Geo. Orrok, Amer. Soc. of Mech. Eng., 1911.

[8] **The Effect of Surface Film on Heat Transfer.**—Heat transfer in ordinary cases is often confused with the thermal conductivity of the material through which the heat has to pass. For example it is said in the proceedings of an important association that, since copper is seven times as good in conductivity as iron is, replacing iron tubes with copper ones in an evaporator or heater should increase the capacity seven times. Actually it increases the capacity only about 10 per cent.

This indicates that there is some factor in heat transfer for which one must look further. If one considers a plate of copper 1 ft. square and 0.065 in. thick the usual values for the thermal conductivity of copper would indicate that there should flow through the plate, per degree F. difference of temperature on the two sides of the plate per hour, about 40,000 B.t.u. If this were an iron plate, about 6500 B.t.u. should pass through under the same conditions. In actual practice the values taken are usually from 250 to 300 B.t.u. though it is possible that the practical value may go up to as much as 1000 under unusual conditions.

Evidently there is some resistance other than the resistance of the metal itself. The usual explanation is that a *film of almost stagnant liquid* is present on the surface of the metal. Water and most liquids are very poor conductors of heat. It is found, for example, that if a film of water 0.01 in. thick is on one side of a copper plate it will decrease the rate of heat transfer from 40,000 to 300 B.t.u. Evidently then the problem in commercial work is to get heat, not through the metal wall, but through the thin films of stagnant liquids or gas. It is necessary, therefore, to know what factors affect the thickness of these films and how it is possible to predict their resistance.

Consider a tube with water on the inside and steam on the outside, and also the resistance that is offered to the flow of heat from the tube to the liquid. It is found that the greater the velocity of the liquid the thinner the stagnant film and that the greater the viscosity the thicker the film. For a film of given thickness the narrower tubes or denser liquids result in higher rates of heat transfer. These factors can be calculated numerically for a few limited cases which arise in practice. For the great bulk of practical cases it is possible to predict only in part what effect a given change in design may have.

formerly in purging the condenser and in preventing air leakage and air accumulation into the system.

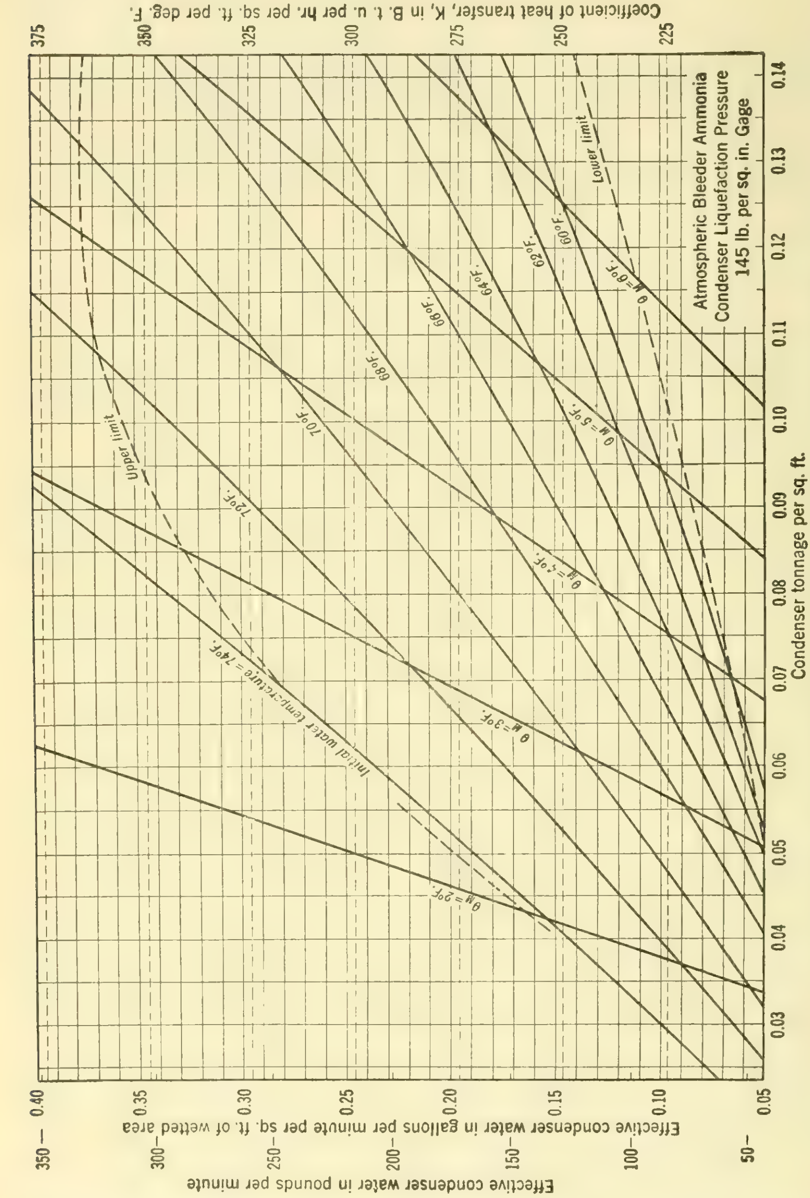


Fig. 160.—Heat Transfer—Atmospheric Type of Bleeder Condenser.

Orrok's tests also brought out that the value of k is dependent also on the mean temperature difference (t_m). In tests by the author on a

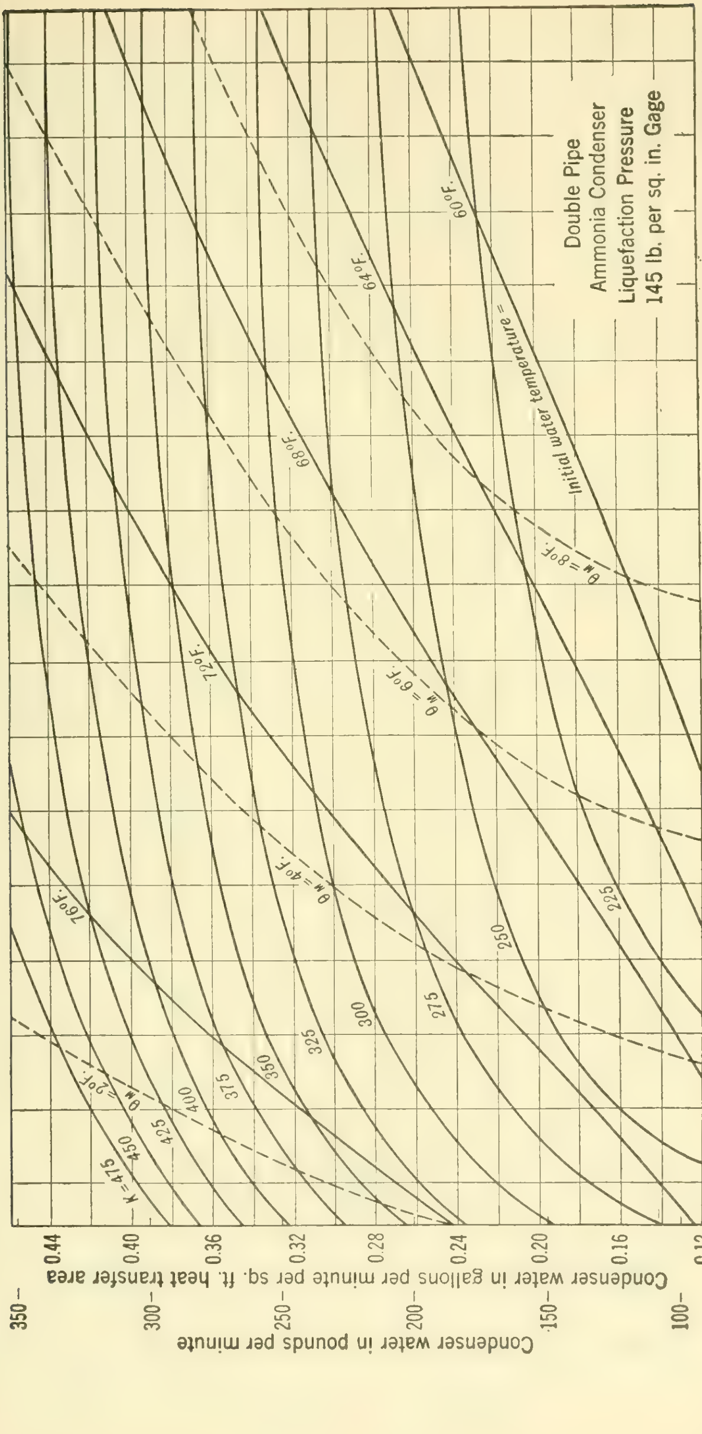


FIG. 161.—Heat Transfer—Double-pipe Ammonia Condenser.

drip type condenser, the value of k was found to increase with a decrease in the mean temperature difference. Showering the condenser with water had the effect of making each pipe more effective, and of decreasing the difference in the amount of liquefaction between the upper and lower pipes, thereby making the lower pipes assume some of the load.[9] In the double pipe and the shell and tube condenser the velocity of the condensing water has a great effect on the value of k, but the atmospheric types do not appear to be affected by showering more or less water on them as far as the water flow is concerned.

In regard to the value of k for the cooling of air, water, brine, oil, milk, etc., the cleanliness of the surface, the velocity of the refrigerant and the commodity being cooled—in so far as it affects the surface through which the heat is passing and kind of fluid being cooled—are the factors which affect it. Frostation varies in amount, the piping design permits more or less surface to be non-effective and oils and dirt are permitted to accumulate inside of the evaporator surface. Only the shell and the tube brine cooler has reasonably standard conditions, although the ice plant piping under flooded conditions is uniform in its performance.

The engineer is interested in the working value of k, one that can be applied to the formula in the solution of problems, and a few of these are as follows:

	B.t.u.
Can ice making piping:	
Old style feed, non-flooded	12 to 15
Flooded	20 to 30
Ammonia condensers:	
Submerged (obsolete except for CO_2)	30 to 40
Atmospheric, gas entering at top	60 to 65
Atmospheric, drip or bleeder	125 to 200
Flooded	125 to 150
Shell and tube	150 to 300
Double pipe	150 to 250
Baudalot coolers, counter flow, atmospheric type:	
Milk coolers	75
Cream coolers	60
Oil coolers	10
Water coolers, for direct expansion	60
Water coolers, for flooded	80
Brine coolers:	
Shell and tube	90 to 100
Double pipe	150 to 300

[9] The drip pipe condenser was one of 2 in. pipes, 12 high, 20 ft. long.

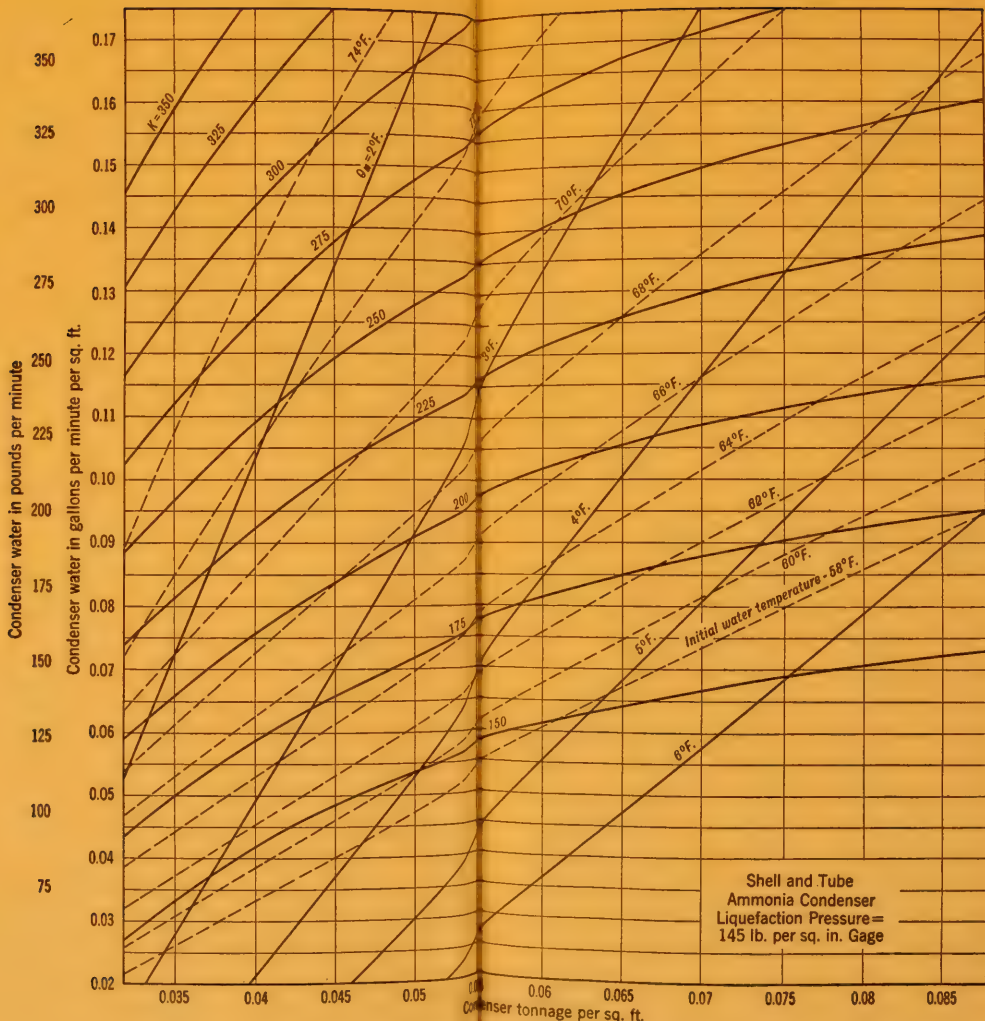


Fig. 162.—Heat Transfer—Shell and Tube Condenser.

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Cooling coils:	
Brine to unagitated air	2 to $2\frac{1}{4}$
Direct expansion	$1\frac{1}{2}$ to 2
Water cooler, shell and coil	15 to 25
Liquid ammonia cooler, shell and coil accumulator.	45
Air dehydrator:	
Shell and coil (brine in coil) 1st coil	5.0
Shell and coil (brine in coil) 2d coil	3.0
Double pipe	6 to 7
Superheat remover, shell and tube	15 to 25

The action of forced circulation of the air increases k by an amount varying from $1\frac{1}{2}$ to $2\frac{1}{2}$ times the values for unagitated air whereas frost decreases the value of k. One inch of frost is considered to decrease the value of k 25 per cent.

Condenser Surfaces.—The values usually taken in specifying the size of condensers are as follows, using the value of 260 B.t.u. per ton per minute to be removed by the condensing water. Shell and tube—12 to 15 sq. ft. per ton; shell and coil—16 sq. ft.; double pipe—6 to 10 sq. ft.; and the flooded condenser 5 to 10 sq. ft.

As an example of the method used in dealing with heat transfer the following will be solved:

Problem.—A 100-ton ice-making plant (160 tons of refrigeration) makes use of the shell and tube superheat remover. The discharge ammonia gas is cooled to 125 deg. F. Water enters at 70 and is heated to 150 deg. F. Find the amount of water that can be heated (for can dipping or other purposes) and the surface required of the superheat remover, if the condenser pressure is 200 and the suction 35 lb. per sq. in. abs.

The temperature of discharge from the compressor is 234 deg. F. and the amount of heat removed from the ammonia, per pound, is $725 - 655 = 70$ B.t.u.

The number of pounds of ammonia passing through the superheat remover is

$$\frac{200}{613.6 - 150.9} \times 160 \times 60 = 4150.$$

The heat units per hour to be removed $= 70 \times 4150 = 290{,}500$ B.t.u. The amount of water heated is $\frac{290{,}500}{150 - 70} = 3631$ lb. per hour. Taking the value of k as 20.0 per hour, and the logarithmic mean temperature difference as $\frac{84 - 55}{\log_e \frac{84}{55}} = 68.5$ deg. F. Then $290{,}500 = A \times 20 \times 68.5$

$$A = \frac{290{,}500}{1370} = 212 \text{ sq. ft.}$$

The superheat remover should therefore have a heat transfer surface of 212 sq. ft.

The Mean Temperature Difference.—In calculating the mean temperature difference the same formula is used where the hotter temperature is constant and the colder is varying (the condenser), the colder constant and the hotter varying (the brine cooler) or the hotter and the

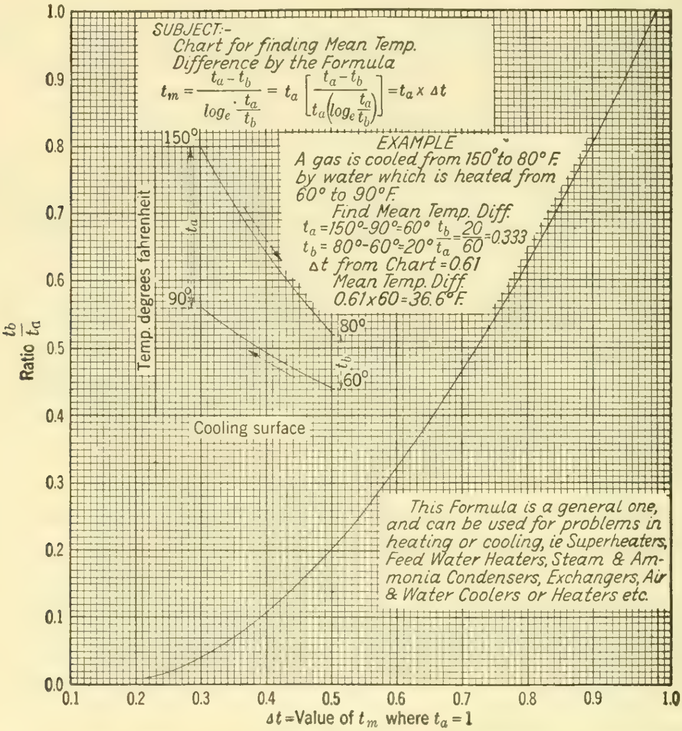

FIG. 163.—Method of Calculating Mean Temperature Difference.

cooler both varying either with parallel or with counter flow of the two fluids. For the first case the formula may be derived as follows:

Referring to Fig. 158, the heat absorbed from a small area dS per unit time is,

$$dQ = CMd\theta$$

where

C = specific heat of the material;

M = weight of the fluid flowing;

$d\theta$ = small change of temperature.

The transfer of heat through the area dS in the same time is

$$dQ = \theta k dS = cM d\theta$$

therefore,

$$\frac{d\theta}{\theta} = \frac{k}{CM} dS,$$

which can be integrated and rearranged to

$$\theta_m = \frac{\theta_a - \theta_b}{\log_e \dfrac{\theta_a}{\theta_b}}.*$$

Values for this expression are shown in the curve in Fig. 163.

$$* \int_b^a \frac{d\theta}{\theta} = \frac{k}{CM} \int_a^b dS$$

Integrating

$$\log_e \frac{\theta_a}{\theta_b} = \frac{kS}{CM}$$

$$CM(\theta_a - \theta_b) = \frac{kS(\theta_a - \theta_b)}{\log_e \dfrac{\theta_a}{\theta_b}} = \Delta Q$$

Therefore

$$\frac{\Delta Q}{kS} = \frac{\theta_a - \theta_b}{\log_e \dfrac{\theta_a}{\theta_b}} = \theta_m$$

Condenser Tests.—Figs. 160, 161 and 162 are taken from Bulletin No. 171 of the Engineering Experiment Station of the University of Illinois by Kratz, Macintire and Gould.

TABLE 49

STANDARD BRINE THICKNESS [CORK INSULATION]

Pipe Size, Inches	Transmission in B.t.u. per Linear Foot per Degree of Difference in Temperature per Hour	Pipe Size, Inches	Transmission in B.t.u. per Linear Foot per Degree of Difference In Temperature per Hour	Pipe Size, Inches	Transmission in B.t.u. per Linear Foot per Degree of Difference in Temperature per Hour
½	0.141	3	0.217	8	0.364
¾	0.147	3½	0.228	9	0.377
1	0.156	4	0.259	10	0.417
1¼	0.161	4½	0.247	12	0.478
1½	0.165	5	0.281	14	0.515
2	0.185	6	0.293	16	0.575
2½	0.202	7	0.354		
SPECIAL THICK BRINE [CORK INSULATION]					
½	0.122	3	0.201	8	0.296
¾	0.132	3½	0.199	9	0.320
1	0.134	4	0.221	10	0.346
1¼	0.143	4½	0.209	12	0.393
1½	0.153	5	0.234	14	0.422
2	0.163	6	0.249	16	0.470
2½	0.184	7	0.273		

ROOMS OR BUILDINGS

Range of Temperatures, Degrees F.	Thicknesses of Corkboard Recommended for				
	Walls, inches	Ceilings, inches	Floors on ground, inches	Floors above ground, inches	Roofs, inches
Below −15	8	8	7	8	9
−15 to −5	7	7	6	7	8
−5 to 10	6	6	5	6	7
10 to 25	5	5	4	5	6
25 to 40	4	4	3	4	5
40 to 50	3	3	2	3	4
50 to 65	2	2	....	2	3
60 and above	....	....	....	....	2

FREEZING TANKS

	Thicknesses of Corkboard Recommended for		
	Bottoms, if placed on foundation laid on ground, inches	Temperatures, Degrees F.	Thickness of Corkboard, inches
Minimum......	5	−20 to −5	8
Preferably.....	6	−5 to +5	6
		5 to 20	5
		20 to 35	4
		35 to 45	3
		45 and above	2

CYLINDRICAL COOLERS, TANKS AND FILTERS FOR COLD LIQUIDS

Range of Temperatures Degrees F.	Thicknesses of Corkboard Recommended for Sides, Top, Inches
Below 0	6
0 to 10	5
10 to 25	4
25 to 45	3
45 to 55	2
55 and above	1½

TABLE 50

[No Void Cork]

Thickness of Sheets, Inches	Number of Sheets in Crate	Measurement of Sheets		Size of Crate		Gross Weight, Pounds
		Dimensions, inches	Board measure, square feet	Dimensions, inches	Contents cubic feet	
1	60	36×12×1	180	38×32×26	18.3	180
$1\frac{1}{2}$	40	36×12×$1\frac{1}{2}$	180	38×32×26	18.3	180
2	28	35×12×2	168	38×30×26	17.1	168
3	20	36×12×3	180	38×32×26	18.3	180
4	16	36×12×4	192	38×34×26	19.3	192

TABLE 51

[Armstrong Cork]

DIMENSIONS AND SHIPPING WEIGHTS

Material	Thickness, Inches	Number of Boards per Crate	Square Feet per Crate	Gross Weight per Crate, Pounds	Gross Weight per Square Foot, Pounds	Net Weight per Square Foot, Pounds
Corkboard	$1\frac{1}{2}$	12	36	52	1.45	1.31
	2	9	27	51	1.89	1.70
	3	6	18	50	2.75	2.47
	4	4	12	45	3.75	3.3
	6	3	9	45	5.00	4.48

All corkboard is made in sheets 12×36 in. The standard size shipping crate (for $1\frac{1}{2}$-in. to 6-in. thick boards) measures 38×21×13 in., the cubical contents being six cubic feet.

As cork is a natural product and hence does not run uniform in weight, all weights shown in the above table are subject to a maximum variation of 10 per cent, over or under.

TABLE 52

Sizes of Cold Storage Doors

[Jamieson]

Size Number	Dimensions inside of Frame or Door "In the Clear"				Wall Opening Required				Cubic Feet, Crated	Approximate Shipping, Weight, Pounds
	Width		Height		Width		Height			
	Ft.	In.	Ft.	In.	Ft.	In.	Ft.	In.		
00	2	6	2	..	3	1	2	5	14	150
0	2	..	3	..	2	7	3	5	15	195
1	2	..	6	..	2	7	6	5	25	295
2	2	6	6	..	3	1	6	5	30	320
3	2	6	6	6	3	1	6	11	32	340
4	3	..	6	..	3	7	6	5	33	350
5	3	..	6	6	3	7	6	11	35	385
6	3	6	6	..	4	1	6	5	37	400
7	3	6	6	6	4	1	6	11	39	435
8	3	..	7	..	3	7	7	5	38	410
9	3	6	7	..	4	1	7	5	42	435
10	4	..	6	..	4	7	6	5	40	435
11	4	..	6	6	4	7	6	11	44	450
13	4	6	6	6	5	1	6	11	48	540
14	4	..	7	..	4	7	7	5	46	530
15	4	6	7	..	5	1	7	5	51	560
16	5	..	6	6	5	7	6	11	51	585
17	5	..	7	..	5	7	7	5	55	600
Sizes of Track Doors										
25	3	6	6	$7\frac{1}{2}$	4	1	7	10	57	510
26	3	6	9	$7\frac{1}{2}$	4	1	10	10	79	700
27	4	..	6	$7\frac{1}{2}$	4	7	7	10	62	550
28	4	..	9	$7\frac{1}{2}$	4	7	10	10	85	780

The "Height in the Clear" for the track doors above are suitable for tracks measuring 7 ft. 2 in. and 10 ft. 2 in., respectively, from finished floor level to top of rail.

CHAPTER VII

REFRIGERANTS

In Gottsche's Die Kälte-Maschinen (1915) there are 27 German firms listed as builders of ammonia compressors, 29 of carbon dioxide and 17 of sulphide dioxide compressors. In Great Britain—due possibly to law but also to the relatively cold condensing water—the carbonic compressor has been much more popular than that using ammonia up to recent years, whereas the sulphur dioxide type has been used relatively very little. Apparently in all countries the tendency is to use the ammonia system in preference to all others. In the United States ammonia has nearly the entire tonnage except for certain localities and kinds of work where the carbonic system has the preference and for small compressors where sulphur dioxide or some of the special refrigerants are coming into use. Viewed impartially, sulphur dioxide[1] has no present excuse for being used, for the charge of the refrigerant and all oil entering the system has to be *anhydrous* or the acid formed with the water will start corrosion. A leak occurring at a joint, for instance, will also start corrosion sufficient to put the pipe into bad shape in a short time. The only successful sulphur dioxide compressor in the United States in capacities up to 10 tons of refrigeration is the one where the machine is very self-contained and hermetically sealed at the factory.

The matter of detection of leaks is a factor of great importance, especially in the fractional tonnage machines where service has to be supplied, and where the initial charge of the refrigerant is from 3 to 5 lb. This factor is of particular moment in the case of certain refrigerants which, like methyl chloride, do not react chemically with any convenient "indicator" as ammonia and sulphur dioxide do.

The factors influencing the choice of a refrigerant are several in number, as follows: A good refrigerant should be non-corrosive to the piping, the compressor, gages, etc.; it should be harmless to people and commodities; it should be of nominal condenser pressures and yet

[1] According to W. S. Douglas, of Wm. Douglas & Sons, in a letter to Cold Storage, May, 1925, the use of SO_2 in Great Britain is increasing steadily, and he considers it the best refrigerant for small machines on account of the low pressure, the lubricating properties, and the ease with which it can be handled.

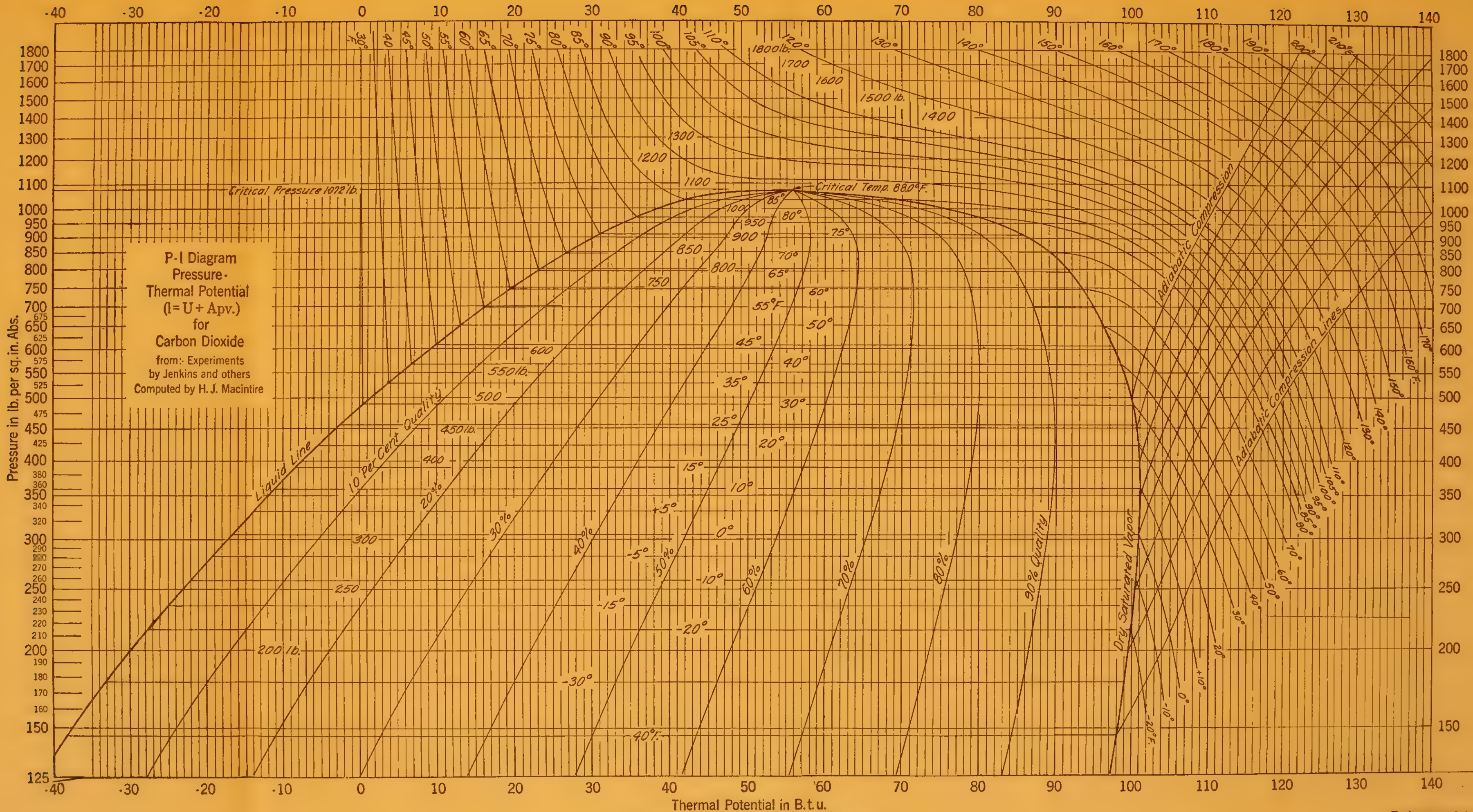

FIG. 164.—P-I Diagram for Carbon Dioxide.

To face page 224.

high suction pressures—300 to 450 lb. per sq. in.—permit a low value of the piston displacement per ton of refrigeration, but because of these heavy unit pressures the cylinder diameter is kept as low as possible by the use of a long stroke, the ratio of the stroke to the diameter being from $3\frac{1}{2}$ to 4. Table 53 gives a good idea of the comparison of the refrigerants, using carbon dioxide as unity in the last column. Table 54 gives values for the thermodynamic properties of carbon dioxide.

TABLE 54

THERMODYNAMIC PROPERTIES OF CARBON DIOXIDE

Temperature, Degrees F.	Pressure, Pounds per Square Inch	Thermal Potential of the Liquid, B.t.u. per Pound	Latent Head of Vaporization	Thermal Potential of Saturated Vapor, B.t.u. per Pound	Density in Cubic Feet per Pound		Entropy	
					Of the liquid	Of the vapor	Of the liquid	Of the vapor
−40	145.87	−38.5	136.5	98.0	0.0143	0.607	−0.0850	0.2400
−35	161.33	−35.8	134.3	98.5	0.0145	0.550	−0.0793	0.2367
−30.0	177.97	−33.1	132.1	99.0	0.0146	0.498	−0.0735	0.2336
−25.0	195.85	−30.4	129.8	99.4	0.0148	0.451	−0.0676	0.2306
−20.0	215.02	−27.7	127.5	99.8	0.0149	0.409	−0.0619	0.2277
−15.0	235.53	−24.9	125.0	100.1	0.0151	0.371	−0.0560	0.2250
−10.0	257.46	−22.1	122.4	100.3	0.0153	0.337	−0.0500	0.2220
− 5.0	280.85	−19.4	120.0	100.6	0.0155	0.307	−0.0440	0.2198
0.0	305.76	−16.7	117.5	100.8	0.0157	0.280	−0.0381	0.2173
5.0	332.2	−14.0	115.0	101.0	0.0159	0.256	−0.0322	0.2151
10.0	360.4	−11.2	112.2	101.0	0.0161	0.235	−0.0264	0.2124
15.0	390.2	− 8.4	109.4	101.0	0.0164	0.216	−0.0204	0.2100
20.0	421.8	− 5.5	106.3	100.8	0.0166	0.199	−0.0144	0.2071
25.0	455.3	− 2.5	103.1	100.6	0.0169	0.182	−0.0083	0.2043
30.0	490.6	+ 0.4	99.7	100.1	0.0172	0.168	−0.0021	0.2012
35.0	528.0	3.5	95.8	99.3	0.0175	0.155	+0.0039	0.1975
40.0	567.3	6.6	91.8	98.4	0.0178	0.143	0.0099	0.1934
45.0	608.9	9.8	87.5	97.3	0.0182	0.132	0.0160	0.1892
50.0	652.7	12.9	83.2	96.1	0.0186	0.122	0.0220	0.1852
55.0	698.8	16.1	78.7	94.8	0.0191	0.112	0.0282	0.1809
60.0	747.4	19.4	74.0	93.4	0.0197	0.093	0.0345	0.1767
65.0	798.6	22.9	68.9	91.8	0.0204	0.085	0.0412	0.1724
70.0	852.5	26.6	62.7	89.3	0.0212	0.077	0.0482	0.1665
75.0	909.3	30.9	54.8	85.7	0.0222	0.070	0.0562	0.1587
80.0	969.3	35.6	44.0	79.6	0.0238	0.063	0.0649	0.1464
85.0	1032.7	41.7	27.5	69.2	0.0266	0.057	0.0761	0.1265
88.0	1072.1	Critical Point						

Ammonia.—Ammonia has nominal unit pressures, never being much more than 185 lb. per sq. in., although the presence of air in the condenser may give a higher value on the gage, or extremely high temperature of the condensing water may send the pressure over 200 lb. The value (Table 55) of the latent heat of vaporization (r) is high—from 550 to 590 B.t.u.—at usual evaporating temperatures, but the gas is extremely noxious to people and has a bad effect on commodities if heavy leaks occur. As leaks are harder to prevent and are more expensive than they would be for steam under similar pressures, the result is a special form of flanged joint (see Chapter IV). Tests[3] seem to indicate that ammonia is an explosive for certain confined mixtures with air from 13.1 to 26.8 per cent by volume, and some very destructive ammonia explosions have occurred from this cause. Ammonia is found to be corrosive to copper and copper compositions if water is present, and therefore steel and iron are used exclusively in the circuit, and the valves and gages are especially constructed with this point in view. Great care is taken against accidents and the possible loss of life. Where liquid gage glasses are used, special automatic shut-off valves are employed to safeguard against the effect of the glass breaking, and remote control stop valves are frequently used in the larger plants to close the mains near the compressor should a cylinder head or other trouble occur to the compressor.

The decomposition of ammonia has been a subject of interest for years. Lowenstein gives as the result of much testing that no decomposition occurs if the ammonia is pure, even with pressures of 300 lb. for a few days at a time. If water dropped on the piston rod this would work into the cylinder. Certain lubricants used for cylinder lubrication were found to decompose in part and the hydrogen component to collect in the condenser. If so-called foul gases are found in the condenser, and good mineral oil is used, it is very nearly positive that these gases are air. If the gases burn it is the ammonia coming out with the air as it is not practical to purge without the loss of ammonia.

Sulphur Dioxide.—Sulphur dioxide has much lower pressures than ammonia but is corrosive to both iron and steel if water is present, even in small amounts. It is very noxious both to people and commodities, although not injurious to *plant life* in small amounts in the air. The latent heat of vaporization (r) is about 170 B.t.u. per lb. and the piston displacement per ton of refrigeration is nearly three times that of ammonia for the same operating conditions. It is not a combustible and there is no danger of explosion from leaks or in the event of accidents to the compressor or to the pipe system. In spite of the natural lubricat-

[3] Arthur Lowenstein, American Society of Brewing Technology, March 29, 1916.

TABLE 55

Saturated Ammonia: Temperature Table

Temperature, Degrees F.	Pressure: Absolute, pounds per inch²	Pressure: Gage, pounds per inch²	Volume of Vapor, Feet³ per Pound	Density of Vapor, Pounds per Foot³	Heat Content: Liquid, B.t.u. per pound	Heat Content: Vapor, B.t.u. per pound	Latent Heat, B.t.u. per Pound	Entropy: Liquid, B.t.u. per pound deg. F.	Entropy: Vapor, B.t.u. per pound deg. F.	Temperature, Degrees F.
t	p	$g.p.$	V	$\frac{1}{V}$	h	H	L	s	S	t
−60	5.55	18.6*	44.73	0.02235	−21.2	589.6	610.8	−0.0517	1.4769	**−60**
−59	5.74	18.2*	43.37	.02306	−20.1	590.0	610.1	− .0490	.4741	−59
−58	5.93	17.8*	42.05	.02378	−19.1	590.4	609.5	− .0464	.4713	−58
−57	6.13	17.4*	40.79	.02452	−18.0	590.8	608.8	− .0438	.4686	−57
−56	6.33	17.0*	39.56	.02528	−17.0	591.2	608.2	− .0412	.4658	−56
−55	6.54	16.6*	38.38	0.02605	−15.9	591.6	607.5	−0.0386	1.4631	**−55**
−54	6.75	16.2*	37.24	.02685	−14.8	592.1	606.9	.0360	.4604	−54
−53	6.97	15.7*	36.15	.02766	−13.8	592.4	606.2	− .0334	.4577	−53
−52	7.20	15.3*	35.09	.02850	−12.7	592.9	605.6	− .0307	.4551	−52
−51	7.43	14.8*	34.06	.02936	−11.7	593.2	604.9	− .0281	.4524	−51
−50	7.67	14.3*	33.08	0.03023	−10.6	593.7	604.3	−0.0256	1.4497	**−50**
−49	7.91	13.8*	32.12	.03113	− 9.6	594.0	603.6	− .0230	.4471	−49
−48	8.16	13.3*	31.20	.03205	− 8.5	594.4	602.9	− .0204	.4445	−48
−47	8.42	12.8*	30.31	.03299	− 7.4	594.9	602.3	− .0179	.4419	−47
−46	8.68	12.2*	29.45	.03395	− 6.4	595.2	601.6	− .0153	.4393	−46
−45	8.95	11.7*	28.62	0.03494	− 5.3	595.6	600.9	−0.0127	1.4368	**−45**
−44	9.23	11.1*	27.82	.03595	− 4.3	596.0	600.3	− .0102	.4342	−44
−43	9.51	10.6*	27.04	.03698	− 3.2	596.4	599.6	− .0076	.4317	−43
−42	9.81	10.0*	26.29	.03804	− 2.1	596.8	598.9	− .0051	.4292	−42
−41	10.10	9.3*	25.56	.03912	− 1.1	597.2	598.3	− .0025	.4267	−41
−40	10.41	8.7*	24.86	0.04022	0.0	597.6	597.6	0.0000	1.4242	**−40**
−39	10.72	8.1*	24.18	.04135	1.1	598.0	596.9	.0025	.4217	−39
−38	11.04	7.4*	23.53	.04251	2.1	598.3	596.2	.0051	.4193	−38
−37	11.37	6.8*	22.89	.04369	3.2	598.7	595.5	.0076	.4169	−37
−36	11.71	6.1*	22.27	.04489	4.3	599.1	594.8	.0101	.4144	−36
−35	12.05	5.4*	21.68	0.04613	5.3	599.5	594.2	0.0126	1.4120	**−35**
−34	12.41	4.7*	21.10	.04739	6.4	599.9	593.5	.0151	.4096	−34
−33	12.77	3.9*	20.54	.04868	7.4	600.2	592.8	.0176	.4072	−33
−32	13.14	3.2*	20.00	.04999	8.5	600.6	592.1	.0201	.4048	−32
−31	13.52	2.4*	19.48	.05134	9.6	601.0	591.4	.0226	.4025	−31
−30	13.90	1.6*	18.97	0.05271	10.7	601.4	590.7	0.0250	1.4001	**−30**
−29	14.30	0.8*	18.48	.05411	11.7	601.7	590.0	.0275	.3978	−29
−28	14.71	0.0	18.00	.05555	12.8	602.1	589.3	.0300	.3955	−28
−27	15.12	0.4	17.54	.05701	13.9	602.5	588.6	.0325	.3932	−27
−26	15.55	0.8	17.09	.05850	14.9	602.8	587.9	.0350	.3909	−26
−25	15.98	1.3	16.66	0.06003	16.0	603.2	587.2	0.0374	1.3886	**−25**
−24	16.42	1.7	16.24	.06158	17.1	603.6	586.5	.0399	.3863	−24
−23	16.88	2.2	15.83	.06317	18.1	603.9	585.8	.0423	.3840	−23
−22	17.34	2.6	15.43	.06479	19.2	604.3	585.1	.0448	.3818	−22
−21	17.81	3.1	15.05	.06644	20.3	604.6	584.3	.0472	.3796	−21
−20	18.30	3.6	14.68	0.06813	21.4	605.0	583.6	0.0497	1.3774	**−20**
−19	18.79	4.1	14.32	.06985	22.4	605.3	582.9	.0521	.3752	−19
−18	19.30	4.6	13.97	.07161	23.5	605.7	582.2	.0545	.3729	−18
−17	19.81	5.1	13.62	.07340	24.6	606.1	581.5	.0570	.3708	−17
−16	20.34	5.6	13.29	.07522	25.6	606.4	580.8	.0594	.3686	−16
−15	20.88	6.2	12.97	0.07709	26.7	606.7	580.0	0.0618	1.3664	**−15**
−14	21.43	6.7	12.66	.07898	27.8	607.1	579.3	.0642	.3643	−14
−13	21.99	7.3	12.36	.08092	28.9	607.5	578.6	.0666	.3621	−13
−12	22.56	7.9	12.06	.08289	30.0	607.8	577.8	.0690	.3600	−12
−11	23.15	8.5	11.78	.08490	31.0	608.1	577.1	.0714	.3579	−11
−10	23.74	9.0	11.50	0.08695	32.1	608.5	576.4	0.0738	1.3558	**−10**

* Inches of mercury below one standard atmosphere (29.92 in.).

TABLE 55—*Continued*

Temperature, Degrees F.	Pressure: Absolute, pounds per inch²	Pressure: Gage, pounds per inch²	Volume of Vapor, Feet³ per Pound	Density of Vapor, Pounds per Foot³	Heat Content: Liquid, B.t.u. per pound	Heat Content: Vapor, B.t.u. per pound	Latent Heat, B.t.u. per Pound	Entropy: Liquid, B.t.u. per pound deg. F.	Entropy: Vapor, B.t.u. per pound deg. F.	Temperature, Degrees F.
t	p	$g.p.$	V	$\frac{1}{V}$	h	H	L	s	S	t
−10	23.74	9.0	11.50	0.08695	32.1	608.5	576.4	0.0738	1.3558	−10
− 9	24.35	9.7	11.23	.08904	33.2	608.8	575.6	.0762	.3537	− 9
− 8	24.97	10.3	10.97	.09117	34.3	609.2	574.9	.0786	.3516	− 8
− 7	25.61	10.9	10.71	.09334	35.4	609.5	574.1	.0809	.3495	− 7
− 6	26.26	11.6	10.47	.09555	36.4	609.8	573.4	.0833	.3474	− 6
− 5	26.92	12.2	10.23	0.09780	37.5	610.1	572.6	0.0857	1.3454	− 5
− 4	27.59	12.9	9.991	.1001	38.6	610.5	571.9	.0880	.3433	− 4
− 3	28.28	13.6	9.763	.1024	39.7	610.8	571.1	.0904	.3413	− 3
− 2	28.98	14.3	9.541	.1048	40.7	611.1	570.4	.0928	.3393	− 2
− 1	29.69	15.0	9.326	.1072	41.8	611.4	569.6	.0951	.3372	− 1
0	30.42	15.7	9.116	0.1097	42.9	611.8	568.9	0.0975	1.3352	0
1	31.16	16.5	8.912	.1122	44.0	612.1	568.1	.0998	.3332	1
2	31.92	17.2	8.714	.1148	45.1	612.4	567.3	.1022	.3312	2
3	32.69	18.0	8.521	.1174	46.2	612.7	566.5	.1045	.3292	3
4	33.47	18.8	8.333	.1200	47.2	613.0	565.8	.1069	.3273	4
5	34.27	19.6	8.150	0.1227	48.3	613.3	565.0	0.1092	1.3253	5
6	35.09	20.4	7.971	.1254	49.4	613.6	564.2	.1115	.3234	6
7	35.92	21.2	7.798	.1282	50.5	613.9	563.4	.1138	.3214	7
8	36.77	22.1	7.629	.1311	51.6	614.3	562.7	.1162	.3195	8
9	37.63	22.9	7.464	.1340	52.7	614.6	561.9	.1185	.3176	9
10	38.51	23.8	7.304	0.1369	53.8	614.9	561.1	0.1208	1.3157	10
11	39.40	24.7	7.148	.1399	54.9	615.2	560.3	.1231	.3137	11
12	40.31	25.6	6.996	.1429	56.0	615.5	559.5	.1254	.3118	12
13	41.24	26.5	6.847	.1460	57.1	615.8	558.7	.1277	.3099	13
14	42.18	27.5	6.703	.1492	58.2	616.1	557.9	.1300	.3081	14
15	43.14	28.4	6.562	0.1524	59.2	616.3	557.1	0.1323	1.3062	15
16	44.12	29.4	6.425	.1556	60.3	616.6	556.3	.1346	.3043	16
17	45.12	30.4	6.291	.1590	61.4	616.9	555.5	.1369	.3025	17
18	46.13	31.4	6.161	.1623	62.5	617.2	554.7	.1392	.3006	18
19	47.16	32.5	6.034	.1657	63.6	617.5	553.9	.1415	.2988	19
20	48.21	33.5	5.910	0.1692	64.7	617.8	553.1	0.1437	1.2969	20
21	49.28	34.6	5.789	.1728	65.8	618.0	552.2	.1460	.2951	21
22	50.36	35.7	5.671	.1763	66.9	618.3	551.4	.1483	.2933	22
23	51.47	36.8	5.556	.1800	68.0	618.6	550.6	.1505	.2915	23
24	52.59	37.9	5.443	.1837	69.1	618.9	549.8	.1528	.2897	24
25	53.73	39.0	5.334	0.1875	70.2	619.1	548.9	0.1551	1.2879	25
26	54.90	40.2	5.227	.1913	71.3	619.4	548.1	.1573	.2861	26
27	56.08	41.4	5.123	.1952	72.4	619.7	547.3	.1596	.2843	27
28	57.28	42.6	5.021	.1992	73.5	619.9	546.4	.1618	.2825	28
29	58.50	43.8	4.922	.2032	74.6	620.2	545.6	.1641	.2808	29
30	59.74	45.0	4.825	0.2073	75.7	620.5	544.8	0.1663	1.2790	30
31	61.00	46.3	4.730	.2114	76.8	620.7	543.9	.1686	.2773	31
32	62.29	47.6	4.637	.2156	77.9	621.0	543.1	.1708	.2755	32
33	63.59	48.9	4.547	.2199	79.0	621.2	542.2	.1730	.2738	33
34	64.91	50.2	4.459	.2243	80.1	621.5	541.4	.1753	.2721	34
35	66.26	51.6	4.373	0.2287	81.2	621.7	540.5	0.1775	1.2704	35
36	67.63	52.9	4.289	.2332	82.3	622.0	539.7	.1797	.2686	36
37	69.02	54.3	4.207	.2377	83.4	622.2	538.8	.1819	.2669	37
38	70.43	55.7	4.126	.2423	84.6	622.5	537.9	.1841	.2652	38
39	71.87	57.2	4.048	.2470	85.7	622.7	537.0	.1863	.2635	39
40	73.32	58.6	3.971	0.2518	86.8	623.0	536.2	0.1885	1.2618	40

TABLE 55—*Continued*

Temperature, Degrees F.	Pressure		Volume of Vapor, Feet³ per Pound	Density of Vapor, Pounds per Foot³	Heat Content		Latent Heat, B.t.u. per Pound	Entropy		Temperature, Degrees F.
	Absolute, pounds per inch²	Gage, pounds per inch²			Liquid, B.t.u. per pound	Vapor, B.t.u. per pound		Liquid, B.t.u. per pound deg. F.	Vapor, B.t.u. per pound deg. F.	
t	p	$g.p.$	V	$\frac{1}{V}$	h	H	L	s	S	t
40	73.32	58.6	3.971	0.2518	86.8	623.0	536.2	0.1885	1.2618	**40**
41	74.80	60.1	3.897	.2566	87.9	623.2	535.3	.1908	.2602	41
42	76.31	61.6	3.823	.2616	89.0	623.4	534.4	.1930	.2585	42
43	77.83	63.1	3.752	.2665	90.1	623.7	533.6	.1952	.2568	43
44	79.38	64.7	3.682	.2716	91.2	623.9	532.7	.1974	.2552	44
45	80.96	66.3	3.614	0.2767	92.3	624.1	531.8	0.1996	1.2535	**45**
46	82.55	67.9	3.547	.2819	93.5	624.4	530.9	.2018	.2519	46
47	84.18	69.5	3.481	.2872	94.6	624.6	530.0	.2040	.2502	47
48	85.82	71.1	3.418	.2926	95.7	624.8	529.1	.2062	.2486	48
49	87.49	72.8	3.355	.2981	96.8	625.0	528.2	.2083	.2469	49
50	89.19	74.5	3.294	0.3036	97.9	625.2	527.3	0.2105	1.2453	**50**
51	90.91	76.2	3.234	.3092	99.1	625.5	526.4	.2127	.2437	51
52	92.66	78.0	3.176	.3149	100.2	625.7	525.5	.2149	.2421	52
53	94.43	79.7	3.119	.3207	101.3	625.9	524.6	.2171	.2405	53
54	96.23	81.5	3.063	.3265	102.4	626.1	523.7	.2192	.2389	54
55	98.06	83.4	3.008	0.3325	103.5	626.3	522.8	0.2214	1.2373	**55**
56	99.91	85.2	2.954	.3385	104.7	626.5	521.8	.2236	.2357	56
57	101.8	87.1	2.902	.3446	105.8	626.7	520.9	.2257	.2341	57
58	103.7	89.0	2.851	.3508	106.9	626.9	520.0	.2279	.2325	58
59	105.6	90.9	2.800	.3571	108.1	627.1	519.0	.2301	.2310	59
60	107.6	92.9	2.751	0.3635	109.2	627.3	518.1	0.2322	1.2294	**60**
61	109.6	94.9	2.703	.3700	110.3	627.5	517.2	.2344	.2278	61
62	111.6	96.9	2.656	.3765	111.5	627.7	516.2	.2365	.2262	62
63	113.6	98.9	2.610	.3832	112.6	627.9	515.3	.2387	.2247	63
64	115.7	101.0	2.565	.3899	113.7	628.0	514.3	.2408	.2231	64
65	117.8	103.1	2.520	0.3968	114.8	628.2	513.4	0.2430	1.2216	**65**
66	120.0	105.3	2.477	.4037	116.0	628.4	512.4	.2451	.2201	66
67	122.1	107.4	2.435	.4108	117.1	628.6	511.5	.2473	.2186	67
68	124.3	109.6	2.393	.4179	118.3	628.8	510.5	.2494	.2170	68
69	126.5	111.8	2.352	.4251	119.4	628.9	509.5	.2515	.2155	69
70	128.8	114.1	2.312	0.4325	120.5	629.1	508.6	0.2537	1.2140	**70**
71	131.1	116.4	2.273	.4399	121.7	629.3	507.6	.2558	.2125	71
72	133.4	118.7	2.235	.4474	122.8	629.4	506.6	.2579	.2110	72
73	135.7	121.0	2.197	.4551	124.0	629.6	505.6	.2601	.2095	73
74	138.1	123.4	2.161	.4628	125.1	629.8	504.7	.2622	.2080	74
75	140.5	125.8	2.125	0.4707	126.2	629.9	503.7	0.2643	1.2065	**75**
76	143.0	128.3	2.089	.4786	127.4	630.1	502.7	.2664	.2050	76
77	145.4	130.7	2.055	.4867	128.5	630.2	501.7	.2685	.2035	77
78	147.9	133.2	2.021	.4949	129.7	630.4	500.7	.2706	.2020	78
79	150.5	135.8	1.988	.5031	130.8	630.5	499.7	.2728	.2006	79
80	153.0	138.3	1.955	0.5115	132.0	630.7	498.7	0.2749	1.1991	**80**
81	155.6	140.9	1.923	.5200	133.1	630.8	497.7	.2769	.1976	81
82	158.3	143.6	1.892	.5287	134.3	631.0	496.7	.2791	.1962	82
83	161.0	146.3	1.861	.5374	135.4	631.1	495.7	.2812	.1947	83
84	163.7	149.0	1.831	.5462	136.6	631.3	494.7	.2833	.1933	84
85	166.4	151.7	1.801	0.5552	137.8	631.4	493.6	0.2854	1.1918	**85**

TABLE 55—*Continued*

Temperature, Degrees F.	Pressure: Absolute, pounds per inch²	Pressure: Gage, pounds per inch²	Volume of Vapor, Feet³ per Pound	Density of Vapor, Pounds per Foot³	Heat Content: Liquid, B.t.u. per pound	Heat Content: Vapor, B.t.u. per pound	Latent Heat, B.t.u. per Pound	Entropy: Liquid, B.t.u. per pound deg. F.	Entropy: Vapor, B.t.u. per pound deg. F.	Te… at… De…
t	p	$g.p.$	V	$\frac{1}{V}$	h	H	L	s	S	
85	166.4	151.7	1.801	0.5552	137.8	631.4	493.6	0.2854	1.1918	
86	169.2	154.5	1.772	.5643	138.9	631.5	492.6	.2875	.1904	
87	172.0	157.3	1.744	.5735	140.1	631.7	491.6	.2895	.1889	
88	174.8	160.1	1.716	.5828	141.2	631.8	490.6	.2917	.1875	
89	177.7	163.0	1.688	.5923	142.4	631.9	489.5	.2937	.1860	
90	180.6	165.9	1.661	0.6019	143.5	632.0	488.5	0.2958	1.1846	
91	183.6	168.9	1.635	.6116	144.7	632.1	487.4	.2979	.1832	
92	186.6	171.9	1.609	.6214	145.8	632.2	486.4	.3000	.1818	
93	189.6	174.9	1.584	.6314	147.0	632.3	485.3	.3021	.1804	
94	192.7	178.0	1.559	.6415	148.2	632.5	484.3	.3041	.1789	
95	195.8	181.1	1.534	0.6517	149.4	632.6	483.2	0.3062	1.1775	
96	198.9	184.2	1.510	.6620	150.5	632.6	482.1	.3083	.1761	
97	202.1	187.4	1.487	.6725	151.7	632.8	481.1	.3104	.1747	
98	205.3	190.6	1.464	.6832	152.9	632.9	480.0	.3125	.1733	
99	208.6	193.9	1.441	.6939	154.0	632.9	478.9	.3145	.1719	
100	211.9	197.2	1.419	0.7048	155.2	633.0	477.8	0.3166	1.1705	1
101	215.2	200.5	1.397	.7159	156.4	633.1	476.7	.3187	.1691	1
102	218.6	203.9	1.375	.7270	157.6	633.2	475.6	.3207	.1677	1
103	222.0	207.3	1.354	.7384	158.7	633.3	474.6	.3228	.1663	1
104	225.4	210.7	1.334	.7498	159.9	633.4	473.5	.3248	.1649	1
105	228.9	214.2	1.313	0.7615	161.1	633.4	472.3	0.3269	1.1635	1
106	232.5	217.8	1.293	.7732	162.3	633.5	471.2	.3289	.1621	1
107	236.0	221.3	1.274	.7852	163.5	633.6	470.1	.3310	.1607	1
108	239.7	225.0	1.254	.7972	164.6	633.6	469.0	.3330	.1593	1
109	243.3	228.6	1.235	.8095	165.8	633.7	467.9	.3351	.1580	1
110	247.0	232.3	1.217	0.8219	167.0	633.7	466.7	0.3372	1.1566	1
111	250.8	236.1	1.198	.8344	168.2	633.8	465.6	.3392	.1552	1
112	254.5	239.8	1.180	.8471	169.4	633.8	464.4	.3413	.1538	1
113	258.4	243.7	1.163	.8600	170.6	633.9	463.3	.3433	.1524	11
114	262.2	247.5	1.145	.8730	171.8	633.9	462.1	.3453	.1510	11
115	266.2	251.5	1.128	0.8862	173.0	633.9	460.9	0.3474	1.1497	11
116	270.1	255.4	1.112	.8996	174.2	634.0	459.8	.3495	.1483	11
117	274.1	259.4	1.095	.9132	175.4	634.0	458.6	.3515	.1469	11
118	278.2	263.5	1.079	.9269	176.6	634.0	457.4	.3535	.1455	1
119	282.3	267.6	1.063	.9408	177.8	634.0	456.2	.3556	.1441	11
120	286.4	271.7	1.047	0.9549	179.0	634.0	455.0	0.3576	1.1427	12
121	290.6	275.9	1.032	.9692	180.2	634.0	453.8	.3597	.1414	12
122	294.8	280.1	1.017	.9837	181.4	634.0	452.6	.3618	.1400	12
123	299.1	284.4	1.002	.9983	182.6	634.0	451.4	.3638	.1386	12
124	303.4	288.7	0.987	1.0132	183.9	634.0	450.1	.3659	.1372	12
125	307.8	293.1	0.973	1.028	185.1	634.0	448.9	0.3679	1.1358	12

TABLE 55a

Saturated Ammonia: Absolute Pressure Table

P sure e) nds r h²	Temperature, Degrees F.	Volume of Vapor, Feet³ per Pound	Density of Vapor, Pounds per Foot³	Heat Content: Liquid, B.t.u. per pound	Heat Content: Vapor, B.t.u. per pound	Latent Heat, B.t.u. per Pound	Entropy: Liquid, B.t.u. per pound deg. F.	Entropy: Evap., B.t.u. per pound deg. F.	Entropy: Vapor, B.t.u. per pound deg. F.	Pressure (Absolute), Pounds per Inch²
	t	V	$\frac{1}{V}$	h	H	L	s	$\frac{L}{T}$	S	p
.0	−63.11	49.31	0.02029	−24.5	588.3	612.8	−0.0599	1.5456	1.4857	**5.0**
.5	−60.27	45.11	.02217	−21.5	589.5	611.0	− .0524	.5301	.4777	5.5
.0	−57.64	41.59	.02405	−18.7	590.6	609.3	− .0455	.5158	.4703	6.0
.5	−55.18	38.59	.02591	−16.1	591.6	607.7	− .0390	.5026	.4636	6.5
.0	−52.88	36.01	.02777	−13.7	592.5	606.2	− .0330	.4904	.4574	7.0
.5	−50.70	33.77	0.02962	−11.3	593.4	604.7	−0.0274	1.4790	1.4516	**7.5**
.0	−48.64	31.79	.03146	− 9.2	594.2	603.4	− .0221	.4683	.4462	8.0
.5	−46.69	30.04	.03329	− 7.1	595.0	602.1	− .0171	.4582	.4411	8.5
.0	−44.83	28.48	.03511	− 5.1	595.7	600.8	− .0123	.4486	.4363	9.0
.5	−43.05	27.08	.03693	− 3.2	596.4	599.6	− .0077	.4396	.4319	9.5
.0	−41.34	25.81	0.03874	− 1.4	597.1	598.5	−0.0034	1.4310	1.4276	**10.0**
.5	−39.71	24.66	.04055	+ 0.3	597.7	597.4	+ .0007	.4228	.4235	10.5
.0	−38.14	23.61	.04235	2.0	598.3	596.3	.0047	.4149	.4196	11.0
.5	−36.62	22.65	.04414	3.6	598.9	595.3	.0085	.4074	.4159	11.5
.0	−35.16	21.77	.04593	5.1	599.4	594.3	.0122	.4002	.4124	12.0
5	−33.74	20.96	0.04772	6.7	600.0	593.3	0.0157	1.3933	1.4090	**12.5**
.0	−32.37	20.20	.04950	8.1	600.5	592.4	.0191	.3866	.4057	13.0
.5	−31.05	19.50	.05128	9.6	601.0	591.4	.0225	.3801	.4026	13.5
.0	−29.76	18.85	.05305	10.9	601.4	590.5	.0257	.3739	.3996	14.0
.5	−28.51	18.24	.05482	12.2	601.9	589.7	.0288	.3679	.3967	14.5
.0	−27.29	17.67	0.05658	13.6	602.4	588.8	0.0318	1.3620	1.3938	**15.0**
.5	−26.11	17.14	.05834	14.8	602.8	588.0	.0347	.3564	.3911	15.5
.0	−24.95	16.64	.06010	16.0	603.2	587.2	.0375	.3510	.3885	16.0
.5	−23.83	16.17	.06186	17.2	603.6	586.4	.0403	.3456	.3859	16.5
0	−22.73	15.72	.06361	18.4	604.0	585.6	.0430	.3405	.3835	17.0
7.5	−21.66	15.30	0.06535	19.6	604.4	584.8	0.0456	1.3354	1.3810	**17.5**
8.0	−20.61	14.90	.06710	20.7	604.8	584.1	.0482	.3305	.3787	18.0
8.5	−19.59	14.53	.06884	21.8	605.1	583.3	.0507	.3258	.3765	18.5
9.0	−18.58	14.17	.07058	22.9	605.5	582.6	.0531	.3211	.3742	19.0
9.5	−17.60	13.83	.07232	23.9	605.8	581.9	.0555	.3166	.3721	19.5
0.0	−16.64	13.50	0.07405	25.0	606.2	581.2	0.0578	1.3122	1.3700	**20.0**
0.5	−15.70	13.20	.07578	26.0	606.5	580.5	.0601	.3078	.3679	20.5
1.0	−14.78	12.90	.07751	27.0	606.8	579.8	.0623	.3036	.3659	21.0
1.5	−13.87	12.62	.07924	27.9	607.1	579.2	.0645	.2995	.3640	21.5
2.0	−12.98	12.35	.08096	28.9	607.4	578.5	.0666	.2955	.3621	22.0
2.5	−12.11	12.09	0.08268	29.8	607.7	577.9	0.0687	1.2915	1.3602	**22.5**
3.0	−11.25	11.85	.08440	30.8	608.1	577.3	.0708	.2876	.3584	23.0
3.5	−10.41	11.61	.08612	31.7	608.3	576.6	.0728	.2838	.3566	23.5
4.0	− 9.58	11.39	.08783	32.6	608.6	576.0	.0748	.2801	.3549	24.0
4.5	− 8.76	11.17	.08955	33.5	608.9	575.4	.0768	.2764	.3532	24.5
5.0	− 7.96	10.96	0.09126	34.3	609.1	574.8	0.0787	1.2728	1.3515	**25.0**
25.5	− 7.17	10.76	.09297	35.2	609.4	574.2	.0805	.2693	.3498	25.5
26.0	− 6.39	10.56	.09468	36.0	609.7	573.7	.0824	.2658	.3482	26.0
26.5	− 5.63	10.38	.09638	36.8	609.9	573.1	.0842	.2625	.3467	26.5
27.0	− 4.87	10.20	.09809	37.7	610.2	572.5	.0860	.2591	.3451	27.0
27.5	− 4.13	10.02	0.09979	38.4	610.4	572.0	0.0878	1.2558	1.3436	**27.5**
28.0	− 3.40	9.853	.1015	39.3	610.7	571.4	.0895	.2526	.3421	28.0
28.5	− 2.68	9.691	.1032	40.0	610.9	570.9	.0912	.2494	.3406	28.5
29.0	− 1.97	9.534	.1049	40.8	611.1	570.3	.0929	.2463	.3392	29.0
29.5	− 1.27	9.383	.1066	41.6	611.4	569.8	.0945	.2433	.3378	29.5
30.0	− 0.57	9.236	0.1083	42.3	611.6	569.3	0.0962	1.2402	1.3364	**30.0**

TABLE 55a—*Continued*

Pressure (Absolute) Pounds per Inch²	Temperature, Degrees F.	Volume of Vapor, Feet³ per Pound	Density of Vapor, Pounds per Foot³	Heat Content		Latent Heat, B.t.u. per Pound	Entropy			Pressure (Absolute), Pounds per Inch²
				Liquid, B.t.u. per pound	Vapor, B.t.u. per pound		Liquid, B.t.u. per pound deg. F.	Evap., B.t.u. per pound deg. F.	Vapor, B.t.u. per pound deg. F.	
p	t	V	$\frac{1}{V}$	h	H	L	s	$\frac{L}{T}$	S	p
30	−0.57	9.236	0.1083	42.3	611.6	569.3	0.0962	1.2402	1.3364	**30**
31	+0.79	8.955	.1117	43.8	612.0	568.2	.0993	.2343	.3336	31
32	2.11	8.693	.1150	45.2	612.4	567.2	.1024	.2286	.3310	32
33	3.40	8.445	.1184	46.6	612.8	566.2	.1055	.2230	.3285	33
34	4.66	8.211	.1218	48.0	613.2	565.2	.1084	.2176	.3260	34
35	5.89	7.991	0.1251	49.3	613.6	564.3	0.1113	1.2123	1.3236	**35**
36	7.09	7.782	.1285	50.6	614.0	563.4	.1141	.2072	.3213	36
37	8.27	7.584	.1319	51.9	614.3	562.4	.1168	.2022	.3190	37
38	9.42	7.396	.1352	53.2	614.7	561.5	.1195	.1973	.3168	38
39	10.55	7.217	.1386	54.4	615.0	560.6	.1221	.1925	.3146	39
40	11.66	7.047	0.1419	55.6	615.4	559.8	0.1246	1.1879	1.3125	**40**
41	12.74	6.885	.1452	56.8	615.7	558.9	.1271	.1833	.3104	41
42	13.81	6.731	.1486	57.9	616.0	558.1	.1296	.1788	.3084	42
43	14.85	6.583	.1519	59.1	616.3	557.2	.1320	.1745	.3065	43
44	15.88	6.442	.1552	60.2	616.6	556.4	.1343	.1703	.3046	44
45	16.88	6.307	0.1586	61.3	616.9	555.6	0.1366	1.1661	1.3027	**45**
46	17.87	6.177	.1619	62.4	617.2	554.8	.1389	.1620	.3009	46
47	18.84	6.053	.1652	63.4	617.4	554.0	.1411	.1580	.2991	47
48	19.80	5.934	.1685	64.5	617.7	553.2	.1433	.1540	.2973	48
49	20.74	5.820	.1718	65.5	618.0	552.5	.1454	.1502	.2956	49
50	21.67	5.710	0.1751	66.5	618.2	551.7	0.1475	1.1464	1.2939	**50**
51	22.58	5.604	.1785	67.5	618.5	551.0	.1496	.1427	.2923	51
52	23.48	5.502	.1818	68.5	618.7	550.2	.1516	.1390	.2906	52
53	24.36	5.404	.1851	69.5	619.0	549.5	.1536	.1354	.2890	53
54	25.23	5.309	.1884	70.4	619.2	548.8	.1556	.1319	.2875	54
55	26.09	5.218	0.1917	71.4	619.4	548.0	0.1575	1.1284	1.2859	**55**
56	26.94	5.129	.1950	72.3	619.7	547.4	.1594	.1250	.2844	56
57	27.77	5.044	.1983	73.3	619.9	546.6	.1613	.1217	.2830	57
58	28.59	4.962	.2015	74.2	620.1	545.9	.1631	.1184	.2815	58
59	29.41	4.882	.2048	75.0	620.3	545.3	.1650	.1151	.2801	59
60	30.21	4.805	0.2081	75.9	620.5	544.6	0.1668	1.1119	1.2787	**60**
61	31.00	4.730	.2114	76.8	620.7	543.9	.1685	.1088	.2773	61
62	31.78	4.658	.2147	77.7	620.9	543.2	.1703	.1056	.2759	62
63	32.55	4.588	.2180	78.5	621.1	542.6	.1720	.1026	.2746	63
64	33.31	4.519	.2213	79.4	621.3	541.9	.1737	.0996	.2733	64
65	34.06	4.453	0.2245	80.2	621.5	541.3	0.1754	1.0966	1.2720	**65**
66	34.81	4.389	.2278	81.0	621.7	540.7	.1770	.0937	.2707	66
67	35.54	4.327	.2311	81.8	621.9	540.1	.1787	.0907	.2694	67
78	36.27	4.267	.2344	82.6	622.0	539.4	.1803	.0879	.2682	68
69	36.99	4.208	.2377	83.4	622.2	538.8	.1819	.0851	.2670	69
70	37.70	4.151	0.2409	84.2	622.4	538.2	0.1835	1.0823	1.2658	**70**
71	38.40	4.095	.2442	85.0	622.6	537.6	.1850	.0795	.2645	71
72	39.09	4.041	.2475	85.8	622.8	537.0	.1866	.0768	.2634	72
73	39.78	3.988	.2507	86.5	622.9	536.4	.1881	.0741	.2622	73
74	40.46	3.937	.2540	87.3	623.1	535.8	.1896	.0715	.2611	74
75	41.13	3.887	0.2573	88.0	623.2	535.2	0.1910	1.0689	1.2599	**75**
76	41.80	3.838	.2606	88.8	623.4	534.6	.1925	.0663	.2588	76
77	42.46	3.790	.2638	89.5	623.5	534.0	.1940	.0637	.2577	77
78	43.11	3.744	.2671	90.2	623.7	533.5	.1954	.0612	.2566	78
79	43.76	3.699	.2704	90.9	623.8	532.9	.1968	.0587	.2555	79
80	44.40	3.655	0.2736	91.7	624.0	532.3	0.1982	1.0563	1.2545	**80**

TABLE 55a—*Continued*

Pressure (Absolute) Pounds per Inch²	Temperature, Degrees F.	Volume of Vapor, Feet³ per Pound	Density of Vapor, Pounds per Foot³	Heat Content		Latent Heat, B.t.u. per Pound	Entropy			Pressure (Absolute), Pounds per Inch²
				Liquid, B.t.u. per pound	Vapor, B.t.u. per pound		Liquid, B.t.u. per pound deg. F.	Evap., B.t.u. per pound deg. F.	Vapor, B.t.u. per pound deg. F.	
p	t	V	$\frac{1}{V}$	h	H	L	s	$\frac{L}{T}$	S	p
80	44.40	3.655	0.2736	91.7	624.0	532.3	0.1982	1.0563	1.2545	**80**
81	45.03	3.612	.2769	92.4	624.1	531.7	.1996	.0538	.2534	81
82	45.66	3.570	.2801	93.1	624.3	531.2	.2010	.0514	.2524	82
83	46.28	3.528	.2834	93.8	624.4	530.6	.2024	.0490	.2514	83
84	46.89	3.488	.2867	94.5	624.6	530.1	.2037	.0467	.2504	84
85	47.50	3.449	0.2899	95.1	624.7	529.6	0.2051	1.0443	1.2494	**85**
86	48.11	3.411	.2932	95.8	624.8	529.0	.2064	.0420	.2484	86
87	48.71	3.373	.2964	96.5	625.0	528.5	.2077	.0397	.2474	87
88	49.30	3.337	.2997	97.2	625.1	527.9	.2090	.0375	.2465	88
89	49.89	3.301	.3030	97.8	625.2	527.4	.2103	.0352	.2455	89
90	50.47	3.266	0.3062	98.4	625.3	526.9	0.2115	1.0330	1.2445	**90**
91	51.05	3.231	.3095	99.1	625.5	526.4	.2128	.0308	.2436	91
92	51.62	3.198	.3127	99.8	625.6	525.8	.2141	.0286	.2427	92
93	52.19	3.165	.3160	100.4	625.7	525.3	.2153	.0265	.2418	93
94	52.76	3.132	.3192	101.0	625.8	524.8	.2165	.0243	.2408	94
95	53.32	3.101	0.3225	101.6	625.9	524.3	0.2177	1.0222	1.2399	**95**
96	53.87	3.070	.3258	102.3	626.1	523.8	.2190	.0201	.2391	96
97	54.42	3.039	.3290	102.9	626.2	523.3	.2201	.0181	.2382	97
98	54.97	3.010	.3323	103.5	626.3	522.8	.2213	.0160	.2373	98
99	55.51	2.980	.3355	104.1	626.4	522.3	.2225	.0140	.2365	99
100	56.05	2.952	0.3388	104.7	626.5	521.8	0.2237	1.0119	1.2356	**100**
102	57.11	2.896	.3453	105.9	626.7	520.8	.2260	.0079	.2339	102
104	58.16	2.843	.3518	107.1	626.9	519.8	.2282	.0041	.2323	104
106	59.19	2.791	.3583	108.3	627.1	518.8	.2305	1.0002	.2307	106
108	60.21	2.741	.3648	109.4	627.3	517.9	.2327	0.9964	.2291	108
110	61.21	2.693	0.3713	110.5	627.5	517.0	0.2348	0.9927	1.2275	**110**
112	62.20	2.647	.3778	111.7	627.7	516.0	.2369	.9890	.2259	112
114	63.17	2.602	.3843	112.8	627.9	515.1	.2390	.9854	.2244	114
116	64.13	2.559	.3909	113.9	628.1	514.2	.2411	.9819	.2230	116
118	65.08	2.517	.3974	114.9	628.2	513.3	.2431	.9784	.2215	118
120	66.02	2.476	0.4039	116.0	628.4	512.4	0.2452	0.9749	1.2201	**120**
122	66.94	2.437	.4104	117.1	628.6	511.5	.2471	.9715	.2186	122
124	67.86	2.399	.4169	118.1	628.7	510.6	.2491	.9682	.2173	124
126	68.76	2.362	.4234	119.1	628.9	509.8	.2510	.9649	.2159	126
128	69.65	2.326	.4299	120.1	629.0	508.9	.2529	.9616	.2145	128
130	70.53	2.291	0.4364	121.1	629.2	508.1	0.2548	0.9584	1.2132	**130**
132	71.40	2.258	.4429	122.1	629.3	507.2	.2567	.9552	.2119	132
134	72.26	2.225	.4494	123.1	629.5	506.4	.2585	.9521	.2106	134
136	73.11	2.193	.4559	124.1	629.6	505.5	.2603	.9490	.2093	136
138	73.95	2.162	.4624	125.1	629.8	504.7	.2621	.9460	.2081	138
140	74.79	2.132	0.4690	126.0	629.9	503.9	0.2638	0.9430	1.2068	**140**
142	75.61	2.103	.4755	126.9	630.0	503.1	.2656	.9400	.2056	142
144	76.42	2.075	.4820	127.9	630.2	502.3	.2673	.9371	.2044	144
146	77.23	2.047	.4885	128.8	630.3	501.5	.2690	.9342	.2032	146
148	78.03	2.020	.4951	129.7	630.4	500.7	.2707	.9313	.2020	148
150	78.81	1.994	0.5016	130.6	630.5	499.9	0.2724	0.9285	1.2009	**150**

TABLE 55*a*—*Continued*

Pressure (Absolute) Pounds per Inch²	Temperature, Degrees F.	Volume of Vapor, Feet³ per Pound	Density of Vapor, Pounds per Foot³	Heat Content		Latent Heat, B.t.u. per Pound	Entropy			Pressure (Absolute), Pounds per Inch²
				Liquid, B.t.u. per pound	Vapor, B.t.u. per pound		Liquid, B.t.u. per pound deg. F.	Evap., B.t.u. per pound deg. F.	Vapor, B.t.u. per pound deg. F.	
p	t	V	$\frac{1}{V}$	h	H	L	s	$\frac{L}{T}$	S	p
150	78.81	1.994	0.5016	130.6	630.5	499.9	0.2724	0.9285	1.2009	**150**
152	79.60	1.968	.5081	131.5	630.6	499.1	.2740	.9257	.1997	152
154	80.37	1.943	.5147	132.4	630.7	498.3	.2756	.9229	.1985	154
156	81.13	1.919	.5212	133.3	630.9	497.6	.2772	.9202	.1974	156
158	81.89	1.895	.5277	134.2	631.0	496.8	.2788	.9175	.1963	158
160	82.64	1.872	0.5343	135.0	631.1	496.1	0.2804	0.9148	1.1952	**160**
162	83.39	1.849	.5408	135.9	631.2	495.3	.2820	.9122	.1942	162
164	84.12	1.827	.5473	136.8	631.3	494.5	.2835	.9096	.1931	164
166	84.85	1.805	.5539	137.6	631.4	493.8	.2850	.9070	.1920	166
168	85.57	1.784	.5604	138.4	631.5	493.1	.2866	.9044	.1910	168
170	86.29	1.764	0.5670	139.3	631.6	492.3	0.2881	0.9019	1.1900	**170**
172	87.00	1.744	.5735	140.1	631.7	491.6	.2895	.8994	.1889	172
174	87.71	1.724	.5801	140.9	631.7	490.8	.2910	.8969	.1879	174
176	88.40	1.705	.5866	141.7	631.8	490.1	.2925	.8944	.1869	176
178	89.10	1.686	.5932	142.5	631.9	489.4	.2939	.8920	.1859	178
180	89.78	1.667	0.5998	143.3	632.0	488.7	0.2954	0.8896	1.1850	**180**
182	90.46	1.649	.6063	144.1	632.1	488.0	.2968	.8872	.1840	182
184	91.14	1.632	.6129	144.8	632.1	487.3	.2982	.8848	.1830	184
186	91.80	1.614	.6195	145.6	632.2	486.6	.2996	.8825	.1821	186
188	92.47	1.597	.6261	146.4	632.3	485.9	.3010	.8801	.1811	188
190	93.13	1.581	0.6326	147.2	632.4	485.2	0.3024	0.8778	1.1802	**190**
192	93.78	1.564	.6392	147.9	632.4	484.5	.3037	.8755	.1792	192
194	94.43	1.548	.6458	148.7	632.5	483.8	.3050	.8733	.1783	194
196	95.07	1.533	.6524	149.5	632.6	483.1	.3064	.8710	.1774	196
198	95.71	1.517	.6590	150.2	632.6	482.4	.3077	.8688	.1765	198
200	96.34	1.502	0.6656	150.9	632.7	481.8	0.3090	0.8666	1.1756	**200**
205	97.90	1.466	.6821	152.7	632.8	480.1	.3122	.8612	.1734	205
210	99.43	1.431	.6986	154.6	633.0	478.4	.3154	.8559	.1713	210
215	100.94	1.398	.7152	156.3	633.1	476.8	.3185	.8507	.1692	215
220	102.42	1.367	.7318	158.0	633.2	475.2	.3216	.8455	.1671	220
225	103.87	1.336	0.7484	159.7	633.3	473.6	0.3246	0.8405	1.1651	**225**
230	105.30	1.307	.7650	161.4	633.4	472.0	.3275	.8356	.1631	230
235	106.71	1.279	.7817	163.1	633.5	470.4	.3304	.8307	.1611	235
240	108.09	1.253	.7984	164.7	633.6	468.9	.3332	.8260	.1592	240
245	109.46	1.227	.8151	166.4	633.7	467.3	.3360	.8213	.1573	245
250	110.80	1.202	0.8319	168.0	633.8	465.8	0.3388	0.8167	1.1555	**250**
255	112.12	1.178	.8487	169.5	633.8	464.3	.3415	.8121	.1536	255
260	113.42	1.155	.8655	171.1	633.9	462.8	.3441	.8077	.1518	260
265	114.71	1.133	.8824	172.6	633.9	461.3	.3468	.8033	.1501	265
270	115.97	1.112	.8993	174.1	633.9	459.8	.3494	.7989	.1483	270
275	117.22	1.091	0.9162	175.6	634.0	458.4	0.3519	0.7947	1.1466	**275**
280	118.45	1.072	.9332	177.1	634.0	456.9	.3545	.7904	.1449	280
285	119.66	1.052	.9502	178.6	634.0	455.4	.3569	.7863	.1432	285
290	120.86	1.034	.9672	180.0	634.0	454.0	.3594	.7821	.1415	290
295	122.05	1.016	.9843	181.5	634.0	452.5	.3618	.7781	.1399	295
300	123.21	0.999	1.0015	182.9	634.0	451.1	0.3642	0.7741	1.1383	**300**

TABLE 55*b*

Properties of Superheated Ammonia

Pressures Are Pounds per Square Inches Absolute

Temperature, Degrees F.	10 —41.34 Degrees		
	V	*H*	*S*
Saturation	*25.81*	*597.1*	*1.4276*
−30	26.58	603.2	1.4420
−20	27.26	608.5	.4542
−10	27.92	613.7	.4659
0	28.58	618.9	1.4773
10	29.24	624.0	.4884
20	29.90	629.1	.4992
30	30.55	634.2	.5097
40	31.20	639.3	.5200
50	31.85	644.4	1.5301
60	32.49	649.5	.5400
70	33.14	654.6	.5497
80	33.78	659.7	.5593
90	34.42	664.8	.5687
100	35.07	670.0	1.5779
110	35.71	675.1	.5870
120	36.35	680.3	.5960
130	36.99	685.4	.6049
140	37.62	690.6	.6136
150	38.26	695.8	1.6222
160	38.90	701.1	.6307
170	39.54	706.3	.6391
180	40.17	711.6	.6474
190	40.81	716.9	.6556
200	41.45	722.2	1.6637

Temperature, Degrees F.	15 —27.29 Degrees		
	V	*H*	*S*
Saturation	*17.67*	*602.4*	*1.3938*
−20	18.01	606.4	1.4031
−10	18.47	611.9	.4154
0	18.92	617.2	1.4272
10	19.37	622.5	.4386
20	19.82	627.8	.4497
30	20.26	633.0	.4604
40	20.70	638.2	.4709
50	21.14	643.4	1.4812
60	21.58	648.5	.4912
70	22.01	653.7	.5011
80	22.44	658.9	.5108
90	22.88	664.0	.5203
100	23.31	669.2	1.5296
110	23.74	674.4	.5388
120	24.17	679.6	.5478
130	24.60	684.8	.5567
140	25.03	690.0	.5655
150	25.46	695.3	1.5742
160	25.88	700.5	.5827
170	26.31	705.8	.5911
180	26.74	711.1	.5995
190	27.16	716.4	.6077
200	27.59	721.7	1.6158
220	28.44	732.4	.6318

Temperature, Degrees F.	20 —16.64 Degrees		
	V	*H*	*S*
Saturation	*13.50*	*606.2*	*1.3700*
−10	13.74	610.0	1.3784
0	14.09	615.5	1.3907
10	14.44	621.0	.4025
20	14.78	626.4	.4138
30	15.11	631.7	.4248
40	15.45	637.0	.4356
50	15.78	642.3	1.4460
60	16.12	647.5	.4562
70	16.45	652.8	.4662
80	16.78	658.0	.4760
90	17.10	663.2	.4856
100	17.43	668.5	1.4950
110	17.76	673.7	.5042
120	18.08	678.9	.5133
130	18.41	684.2	.5223
140	18.73	689.4	.5312
150	19.05	694.7	1.5399
160	19.37	700.0	.5485
170	19.70	705.3	.5569
180	20.02	710.6	.5653
190	20.34	715.9	.5736
200	20.66	721.2	1.5817
220	21.30	732.0	.5978
240	21.94	742.8	.6135

Temperature, Degrees F.	25 —7.96 Degrees		
	V	*H*	*S*
Saturation	*10.96*	*609.1*	*1.3515*
0	11.19	613.8	1.3616
10	11.47	619.4	.3738
20	11.75	625.0	.3855
30	12.03	630.4	.3967
40	12.30	635.8	.4077
50	12.57	641.2	1.4183
60	12.84	646.5	.4287
70	13.11	651.8	.4388
80	13.37	657.1	.4487
90	13.64	662.4	.4584
100	13.90	667.7	1.4679
110	14.17	673.0	.4772
120	14.43	678.2	.4864
130	14.69	683.5	.4954
140	14.95	688.8	.5043
150	15.21	694.1	1.5131
160	15.47	699.4	.5217
170	15.73	704.7	.5303
180	15.99	710.1	.5387
190	16.25	715.4	.5470
200	16.50	720.8	1.5552
220	17.02	731.6	.5713
240	17.53	742.5	.5870
260	18.04	753.4	.6025

Temperature, Degrees F.	30 —0.57 Degree		
	V	*H*	*S*
Saturation	*9.236*	*611.6*	*1.3364*
10	9.492	617.8	1.3497
20	9.731	623.5	.3618
30	9.966	629.1	.3733
40	10.20	634.6	.3845
50	10.43	640.1	1.3953
60	10.65	645.5	.4059
70	10.88	650.9	.4161
80	11.10	656.2	.4261
90	11.33	661.6	.4359
100	11.55	666.9	1.4456
110	11.77	672.2	.4550
120	11.99	677.5	.4642
130	12.21	682.9	.4733
140	12.43	688.2	.4823
150	12.65	693.5	1.4911
160	12.87	698.8	.4998
170	13.08	704.2	.5083
180	13.30	709.6	.5168
190	13.52	714.9	.5251
200	13.73	720.3	1.5334
220	14.16	731.1	.5495
240	14.59	742.0	.5653
260	15.02	753.0	.5808
280	15.45	764.1	.5960

Temperature, Degrees F.	35 5.89 Degrees		
	V	*H*	*S*
Saturation	*7.991*	*613.6*	*1.3236*
10	8.078	616.1	1.3289
20	8.287	622.0	.3413
30	8.493	627.7	.3532
40	8.695	633.4	.3646
50	8.895	638.9	1.3756
60	9.093	644.4	.3863
70	9.289	649.9	.3967
80	9.484	655.3	.4069
90	9.677	660.7	.4168
100	9.869	666.1	1.4265
110	10.06	671.5	.4360
120	10.25	676.8	.4453
130	10.44	682.2	.4545
140	10.63	687.6	.4635
150	10.82	692.9	1.4724
160	11.00	698.3	.4811
170	11.19	703.7	.4897
180	11.38	709.1	.4982
190	11.56	714.5	.5066
200	11.75	719.9	1.5148
220	12.12	730.7	.5311
240	12.49	741.7	.5469
260	12.86	752.7	.5624
280	13.23	763.7	.5776

TABLE 55*b*—*Continued*

Temperature, Degrees F.	40, *11.66 Degrees* V	H	S
Saturation	*7.047*	*615.4*	*1.3125*
20	7.203	620.4	1.3231
30	7.387	626.3	.3353
40	7.568	632.1	.3470
50	7.746	637.8	1.3583
60	7.922	643.4	.3692
70	8.096	648.9	.3797
80	8.268	654.4	.3900
90	8.439	659.9	.4000
100	8.609	665.3	1.4098
110	8.777	670.7	.4194
120	8.945	676.1	.4288
130	9.112	681.5	.4381
140	9.278	686.9	.4471
150	9.444	692.3	1.4561
160	9.609	697.7	.4648
170	9.774	703.1	.4735
180	9.938	708.5	.4820
190	10.10	714.0	.4904
200	10.27	719.4	1.4987
220	10.59	730.3	.5150
240	10.92	741.3	.5309
260	11.24	752.3	.5465
280	11.56	763.4	.5617
300	11.88	774.6	.5766

Temperature, Degrees F.	50, *21.67 Degrees* V	H	S
Saturation	*5.710*	*618.2*	*1.2939*
30	5.838	623.4	1.3046
40	5.988	629.5	3169
50	6.135	635.4	1.3286
60	6.280	641.2	.3399
70	6.423	646.9	.3508
80	6.564	652.6	.3613
90	6.704	658.2	.3716
100	6.843	663.7	1.3816
110	6.980	669.2	.3914
120	7.117	674.7	.4009
130	7.252	680.2	.4103
140	7.387	685.7	4195
150	7.521	691.1	1.4286
160	7.655	696.6	.4374
170	7.788	702.1	.4462
180	7.921	707.5	.4548
190	8.053	713.0	.4633
200	8.185	718.5	1.4716
210	8.317	724.0	.4799
220	8.448	729.4	.4880
240	8.710	740.5	.5040
260	8.970	751.6	.5197
280	9.230	762.7	1.5350
300	9.489	774.0	.5500

Temperature, Degrees F.	60, *30.21 Degrees* V	H	S
Saturation	*4.805*	*620.5*	*1.2787*
40	4.933	626.8	1.2913
50	5.060	632.9	1.3035
60	5.184	639.0	.3152
70	5.307	644.9	.3265
80	5.428	650.7	.3373
90	5.547	656.4	.3479
100	5.665	662.1	1.3581
110	5.781	667.7	.3681
120	5.897	673.3	.3778
130	6.012	678.9	.3873
140	6.126	684.4	.3966
150	6.239	689.9	1.4058
160	6.352	695.5	.4148
170	6.464	701.0	.4236
180	6.576	706.5	.4323
190	6.687	712.0	.4409
200	6.798	717.5	1.4493
210	6.909	723.1	.4576
220	7.019	728.6	.4658
230	7.129	734.1	.4739
240	7.238	739.7	.4819
260	7.457	750.9	1.4976
280	7.675	762.1	.5130
300	7.892	773.3	.5281

Temperature, Degrees F.	75, *41.13 Degrees* V	H	S
Saturation	*3.887*	*623.2*	*1.2599*
50	3.982	629.1	1.2715
60	4.087	635.5	.2839
70	4.189	641.7	.2957
80	4.289	647.7	.3071
90	4.388	653.7	.3180
100	4.485	659.6	1.3286
110	4.581	665.4	.3389
120	4.676	671.1	.3489
130	4.770	676.8	.3586
140	4.863	682.5	.3682
150	4.956	688.1	1.3775
160	5.048	693.7	.3866
170	5.139	699.3	.3956
180	5.230	704.9	.4044
190	5.320	710.5	.4131
200	5.410	716.1	1.4217
210	5.500	721.7	.4301
220	5.589	727.3	.4384
230	5.678	732.9	.4466
240	5.767	738.5	.4546
250	5.855	744.1	1.4625
260	5.943	749.8	.4705
280	6.119	761.1	.4860
300	6.294	772.4	.5011

Temperature, Degrees F.	100, *56.05 Degrees* V	H	S
Saturation	*2.952*	*626.5*	*1.2356*
70	3.068	636.0	1.2539
80	3.149	642.6	.2661
90	3.227	649.0	.2778
100	3.304	655.2	1.2891
110	3.380	661.3	.2999
120	3.454	667.3	.3104
130	3.527	673.3	.3206
140	3.600	679.2	.3305
150	3.672	685.0	1.3401
160	3.743	690.8	.3495
170	3.813	696.6	.3588
180	3.883	702.3	.3678
190	3.952	708.0	.3767
200	4.021	713.7	1.3854
210	4.090	719.4	.3940
220	4.158	725.1	.4024
230	4.226	730.8	.4108
240	4.294	736.5	.4190
250	4.361	742.2	1.4271
260	4.428	747.9	.4350
270	4.495	753.6	.4429
280	4.562	759.4	.4507
290	4.629	765.1	.4584
300	4.695	770.8	1.4660

Temperature, Degrees F.	125, *68.31 Degrees* V	H	S
Saturation	*2.380*	*628.8*	*1.2166*
80	2.461	637.2	1.2322
90	2.528	644.0	.2448
100	2.593	650.7	1.2568
110	2.657	657.1	.2682
120	2.719	663.5	.2792
130	2.780	669.7	.2899
140	2.840	675.8	.3002
150	2.900	681.8	1.3102
160	2.958	687.8	.3199
170	3.016	693.7	.3294
180	3.074	699.6	.3387
190	3.131	705.5	.3478
200	3.187	711.3	1.3567
210	3.243	717.2	.3654
220	3.299	723.0	.3740
230	3.354	728.8	.3825
240	3.409	734.5	.3908
250	3.464	740.3	1.3990
260	3.519	746.1	.4071
270	3.573	751.9	.4151
280	3.627	757.7	.4230
290	3.681	763.5	.4308
300	3.735	769.3	1.4385
320	3.842	780.9	.4536

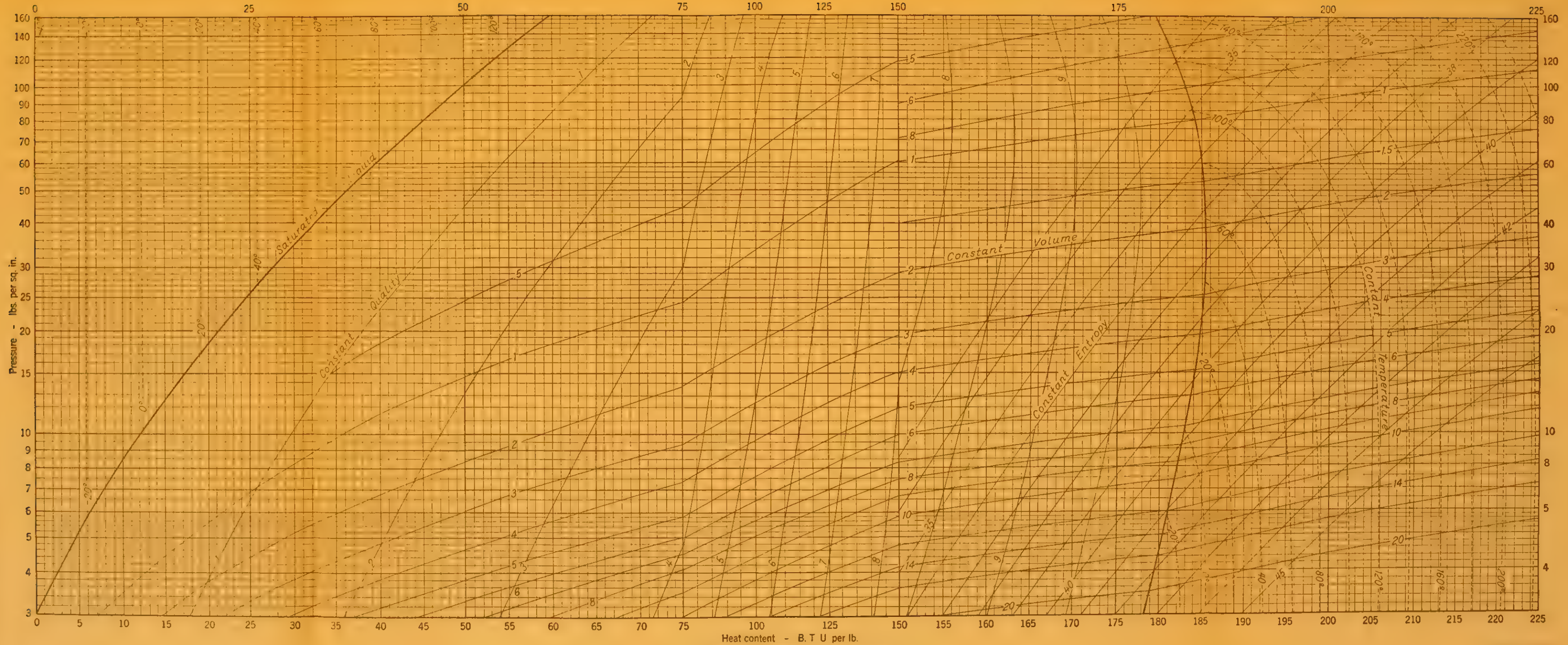

Fig. 166.—P–I Diagram for Sulphur Dioxide.

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TABLE 55*b*—*Continued*

Temperature, Degrees F.	150 78.81 Degrees		
	V	*H*	*S*
Saturation	*1.994*	*630.5*	*1.2009*
90	2.061	638.8	1.2161
100	2.118	645.9	1.2289
110	2.174	652.8	.2410
120	2.228	659.4	.2526
130	2.281	665.9	.2638
140	2.334	672.3	.2745
150	2.385	678.6	1.2849
160	2.435	684.8	.2949
170	2.485	690.9	.3047
180	2.534	696.9	.3142
190	2.583	702.9	.3236
200	2.631	708.9	1.3327
210	2.679	714.8	.3416
220	2.726	720.7	.3504
230	2.773	726.6	.3590
240	2.820	732.5	.3675
250	2.866	738.4	1.3758
260	2.912	744.3	.3840
270	2.958	750.1	.3921
280	3.004	756.0	.4001
290	3.049	761.8	.4079
300	3.095	767.7	1.4157
320	3.185	779.4	.4310
340	3.274	791.2	.4459

Temperature, Degrees F.	200 96.34 Degrees		
	V	*H*	*S*
Saturation	*1.502*	*632.7*	*1.1756*
110	1.567	643.4	1.1947
120	1.612	650.9	.2077
130	1.656	658.1	.2200
140	1.698	665.0	.2317
150	1.740	671.8	1.2429
160	1.780	678.4	.2537
170	1.820	684.9	.2641
180	1.859	691.3	.2742
190	1.897	697.7	.2840
200	1.935	703.9	1.2935
210	1.972	710.1	.3029
220	2.009	716.3	.3120
230	2.046	722.4	.3209
240	2.082	728.4	.3296
250	2.118	734.5	1.3382
260	2.154	740.5	.3467
270	2.189	746.5	.3550
280	2.225	752.5	.3631
290	2.260	758.5	.3712
300	2.295	764.5	1.3791
320	2.364	776.5	.3947
340	2.432	788.5	.4099
360	2.500	800.5	.4247
380	2.568	812.5	.4392

Temperature, Degrees F.	250 110.80 Degrees		
	V	*H*	*S*
Saturation	*1.202*	*633.8*	*1.1555*
120	1.240	641.5	1.1690
130	1.278	649.6	.1827
140	1.316	657.2	.1956
150	1.352	664.6	1.2078
160	1.386	671.8	.2195
170	1.420	678.7	.2306
180	1.453	685.5	.2414
190	1.486	692.2	.2517
200	1.518	698.8	1.2617
210	1.549	705.3	.2715
220	1.580	711.7	.2810
230	1.610	718.0	.2902
240	1.640	724.3	.2993
250	1.670	730.5	1.3081
260	1.699	736.7	.3168
270	1.729	742.9	.3253
280	1.758	749.1	.3337
290	1.786	755.2	.3420
300	1.815	761.3	1.3501
320	1.872	773.5	.3659
340	1.928	785.7	.3814
360	1.983	797.9	.3964
380	2.038	810.1	.4111
400	2.093	822.3	1.4255

ing properties of sulphur dioxide, an oil has to be used and this needs to be absolutely anhydrous or corrosion will develop. Table 56 gives the thermodynamic properties of sulphur dioxide.

Ethyl Chloride.—Ethyl chloride (C_2H_5Cl) is a colorless liquid, slightly lighter in weight than water, which may be stored in moderate pressure containers. It is an anæsthetic but must be inhaled in considerable quantities in order to have an appreciable effect. It is very inflammable and can be exploded by a spark at atmospheric pressures in mixtures of gas in air of not less than 4.9 and not greater than 13.5 per cent by volume. The corresponding mixtures for ammonia and gasoline are 16.5 to 26.8 per cent and 1.5 to 6.4 per cent respectively. It does not corrode iron,[4] nor copper or its alloys, nor aluminum. The condensing pressure is very small—about 27 lb. abs. at 86 deg. F. (Table 57), and the evaporating pressure is usually less than one atmosphere—and the piston displacement per ton of refrigeration is very high, some 30 times that for carbon dioxide and six times that of ammonia. Because of the low pressures and the large volumes to be compressed per unit of refrigeration, the ethyl chloride compressor has been designed as a rotary

[4] Special Report No. 14, Flood Investigation Board, by Jenkin and Shorthose.

TABLE 56

Saturated Sulphur Dioxide: Temperature Table

Temperature, Degrees F.	Pressure (Absolute), Pounds per Inch²	Volume, Foot³-Pounds		Heat Content, B.t.u. per Pound		Latent Heat, B.t.u. per Pound		Entropy		
t	p	Liquid v'	Vapor v''	Liquid q	Vapor i''	Total r	Internal	Liquid s'	$\frac{r}{T}$	Vapor s''
−40	3.136	.010440	22.42	0.00	178.61	178.61	165.57	.0000000	.42562	.42562
−35	3.693	.010486	19.23	1.45	179.27	177.82	164.66	.00334	.41875	.42209
−30	4.331	.010532	16.56	2.93	179.90	176.97	163.70	.00674	.41190	.41864
−25	5.058	.010580	14.31	4.44	180.50	176.06	162.67	.01016	.40509	.41525
−20	5.883	.010627	12.42	5.98	181.07	175.09	161.58	.01366	.39826	.41192
−15	6.814	.010674	10.81	7.56	181.62	174.06	160.44	.01719	.39146	.40865
−10	7.863	.010721	9.44	9.16	182.13	172.97	159.24	.02075	.38469	.40544
− 5	9.038	.010770	8.28	10.79	182.62	171.83	157.29	.02443	.37795	.40228
0	10.35	.010820	7.280	12.44	183.07	170.63	156.70	.02795	.37122	.39917
1	10.63	.010830	7.099	12.79	183.17	170.38	156.44	.02869	.36987	.39856
2	10.91	.010840	6.923	13.12	183.25	170.13	156.17	.02941	.36853	.39794
3	11.20	.010850	6.751	13.45	183.33	169.88	155.90	.03013	.36719	.39732
4	11.50	.010860	6.584	13.78	183.41	169.63	155.63	.03084	.36586	.39670
5	11.81	.010870	6.421	14.11	183.49	169.38	155.36	.03155	.36454	.39609
6	12.12	.010880	6.266	14.45	183.57	169.12	155.09	.03228	.36319	.39547
7	12.43	.010890	6.114	14.79	183.65	168.86	154.81	.03300	.36186	.39486
8	12.75	.010900	5.967	15.13	183.73	168.60	154.54	.03373	.36053	.39426
9	13.08	.010910	5.822	15.46	183.80	168.34	154.26	.03445	.35921	.39366
10	13.42	.010920	5.682	15.80	183.87	168.07	153.98	.03519	.35787	.39306
11	13.77	.010930	5.548	16.14	183.94	167.80	153.70	.03592	.35654	.39246
12	14.12	.010940	5.417	16.48	184.01	167.53	153.42	.03664	.35521	.39185
13	14.48	.010950	5.289	16.81	184.07	167.26	153.13	.03737	.35388	.39125
14	14.84	.010960	5.164	17.15	184.14	166.97	152.84	.03808	.35257	.39065
15	15.21	.010971	5.042	17.49	184.21	166.72	152.55	.03880	.35125	.39005
16	15.59	.010981	4.926	17.84	184.28	166.44	152.26	.03953	.34993	.38946
17	15.98	.010992	4.812	18.18	184.34	166.16	151.97	.04026	.34861	.38887
18	16.37	.011003	4.701	18.52	184.40	165.88	151.68	.04098	.34729	.38827
19	16.77	.011014	4.593	18.86	184.46	165.60	151.38	.04169	.34598	.38767
20	17.18	.011025	4.487	19.20	184.52	165.32	151.08	.04241	.34466	.38707
21	17.60	.011036	4.386	19.55	184.58	165.03	150.78	.04313	.34335	.38648
22	18.03	.011047	4.287	19.90	184.64	164.74	150.48	.04385	.34204	.38589
23	18.46	.011058	4.190	20.24	184.69	164.45	150.18	.04457	.34073	.38530
24	18.89	.011070	4.096	20.58	184.74	164.16	149.88	.04528	.33943	.38471
25	19.34	.011082	3.994	20.92	184.79	163.87	149.57	.04600	.33812	.38412
26	19.80	.011093	3.915	21.26	184.84	163.58	149.27	.04671	.33683	.38354
27	20.26	.011104	3.829	21.61	184.89	163.28	148.97	.04743	.33553	.38296
28	20.73	.011116	3.744	21.96	184.94	162.98	148.66	.04814	.33422	.38237
29	21.21	.011128	3.662	22.30	184.98	162.68	148.34	.04886	.33292	.38178
30	21.70	.011140	3.581	22.64	185.02	162.38	148.02	.04956	.33163	.38119
31	22.20	.011152	3.503	22.98	185.06	162.08	147.72	.05027	.33034	.38061
32	22.71	.011164	3.437	23.33	185.10	161.77	147.41	.05099	.32904	.38003
33	23.23	.011176	3.355	23.68	185.14	161.46	147.09	.05171	.32774	.37945
34	23.75	.011188	3.283	24.03	185.18	161.15	146.77	.05242	.32645	.37887
35	24.28	.011200	3.212	24.38	185.22	160.84	146.45	.05312	.32517	.37829
36	24.82	.011212	3.144	24.72	185.25	160.53	146.13	.05384	.32388	.37772
37	25.39	.011224	3.078	25.07	185.28	160.21	145.81	.05456	.32259	.37715
38	25.95	.011236	3.013	25.42	185.31	159.89	145.48	.05527	.32130	.37657
39	26.52	.011248	2.949	25.77	185.34	159.57	145.15	.05598	.32001	.37599
40	27.10	.011260	2.887	26.12	185.37	159.25	144.82	.05668	.31873	.37541
41	27.69	.011272	2.827	26.47	185.40	158.93	144.55	.05738	.31745	.47483
42	28.29	.011284	2.769	26.81	185.42	158.61	144.17	.05809	.31616	.37425
43	28.90	.011296	2.712	27.16	185.44	158.28	144.84	.05879	.31489	.37368
44	29.52	.011308	2.656	27.51	185.46	157.95	144.50	.05949	.31362	.37311
45	30.15	.011320	2.601	27.86	185.48	157.62	143.16	.06020	.31234	.37254
46	30.79	.011332	2.548	28.21	185.50	157.29	143.83	.06090	.31107	.37197
47	31.44	.011344	2.497	28.56	185.52	156.96	142.49	.06161	.30979	.37140
48	32.10	.011356	2.446	28.92	185.54	156.62	142.15	.06231	.30852	.37083
49	32.77	.011368	2.397	29.27	185.55	156.28	142.81	.06301	.30725	.37026

TABLE 56—*Continued*

Temperature, Degrees F. t	Pressure (Absolute), Pounds per Inch² p	Volume, Foot³-Pounds		Heat Content, B.t.u. per Pound		Latent Heat, B.t.u. per Pound		Entropy		
		Liquid v'	Vapor v''	Liquid q	Vapor i''	Total r	Internal	Liquid s'	$\frac{r}{T}$	Vapor s''
50	33.45	.011380	2.348	29.61	185.56	155.95	141.47	.06370	.30599	.36969
51	34.15	.011392	2.302	29.96	185.57	159.61	141.13	.06439	.30474	.36913
52	34.86	.011404	2.256	30.31	185.58	155.27	140.79	.06509	.30348	.36857
53	35.58	.011416	2.211	30.66	185.59	154.93	140.44	.06578	.30222	.36800
54	36.31	.011428	2.167	31.00	185.59	154.59	140.09	.06646	.30097	.36743
55	37.05	.011440	2.124	31.36	185.60	154.24	139.74	.06715	.29971	.36686
56	37.80	.011452	2.083	31.72	185.61	153.89	139.39	.06785	.29844	.36629
57	38.56	.011464	2.043	32.08	185.62	153.54	139.04	.06854	.29719	.36573
58	39.33	.011476	2.003	32.42	185.61	153.19	138.69	.06923	.29594	.36517
59	40.12	.011488	1.964	32.76	185.60	152.84	138.34	.06992	.29469	.36461
60	40.93	.011500	1.926	33.10	185.59	152.49	137.98	.07060	.29345	.36405
61	41.75	.011512	1.889	33.44	185.58	152.14	137.53	.07128	.29221	.36349
62	42.58	.011524	1.853	33.79	185.57	151.78	137.17	.07196	.29097	.36293
63	43.42	.011536	1.816	34.14	185.56	151.42	136.81	.07265	.28972	.36237
64	44.27	.011548	1.783	34.49	185.55	151.06	136.35	.07333	.28848	.36181
65	45.13	.011560	1.749	34.84	185.54	150.70	136.19	.07401	.28724	.36125
66	46.00	.011572	1.716	35.19	185.53	150.34	135.83	.07469	.28601	.36070
67	46.88	.011585	1.683	35.54	185.52	149.98	135.47	.07535	.28479	.36014
68	47.78	.011598	1.652	35.88	185.50	149.62	135.11	.07602	.28356	.35958
69	48.69	.011611	1.621	36.23	185.48	149.25	134.74	.07769	.28233	.35902
70	49.62	.011626	1.590	36.58	185.46	148.88	134.37	.07736	.28110	.35846
71	50.57	.011639	1.557	36.93	185.44	148.51	134.01	.07804	.27987	.35791
72	51.54	.011652	1.532	37.28	185.42	148.14	133.64	.07871	.27865	.35736
73	52.51	.011666	1.503	37.63	185.40	147.77	133.27	.07937	.27743	.35680
74	53.48	.011680	1.476	37.97	185.37	147.40	132.90	.08003	.27621	.35624
75	54.47	.011693	1.448	38.32	185.34	147.02	132.53	.08070	.27498	.35568
76	55.48	.011706	1.422	38.67	185.31	146.64	132.16	.08135	.27377	.35512
77	56.51	.011719	1.396	39.01	185.27	146.26	131.79	.08201	.27255	.35456
78	57.56	.011732	1.371	39.36	185.24	145.88	131.41	.08268	.27133	.35401
79	58.62	.011746	1.343	39.71	185.21	145.50	131.03	.08336	.27012	.35346
80	59.68	.011760	1.321	40.05	185.17	145.12	130.65	.08399	.26897	.35291
81	60.77	.011773	1.297	40.39	185.13	144.74	130.27	.08462	.26772	.35234
82	61.88	.011786	1.274	40.73	185.09	144.36	129.89	.08525	.26652	.35177
83	62.01	.011800	1.253	41.08	185.05	143.97	129.51	.08589	.26532	.35121
84	64.14	.011814	1.229	41.43	185.01	143.58	129.13	.08653	.26412	.35065
85	65.28	.011828	1.207	41.78	184.97	143.19	128.75	.08718	.26291	.35009
86	66.45	.011841	1.185	42.12	184.92	142.80	128.37	.08783	.26171	.34954
87	67.64	.011854	1.164	42.46	184.87	142.41	127.99	.08847	.26052	.34899
88	68.84	.011868	1.144	42.80	184.82	142.02	127.60	.08910	.25933	.34843
89	70.04	.011882	1.124	43.15	184.77	141.62	127.21	.08974	.25813	.34787
90	71.25	.011896	1.104	43.50	184.72	141.22	126.82	.09038	.25693	.34731
91	72.46	.011909	1.084	43.85	184.67	140.82	126.43	.09102	.25574	.34676
92	73.70	.011923	1.065	44.19	184.61	140.42	125.04	.09165	.25455	.34620
93	74.98	.011937	1.047	44.53	184.55	140.02	125.65	.09227	.25337	.34564
94	76.30	.011951	1.028	44.86	184.49	139.62	125.26	.09389	.25219	.34508
95	77.60	.011965	1.011	45.20	184.43	139.23	124.87	.09349	.25103	.34452
96	79.03	.011979	.9931	45.54	184.37	138.83	124.48	.09411	.24986	.34397
97	80.40	.011993	.9759	45.88	184.31	138.43	124.09	.09472	.24869	.34341
98	81.77	.012008	.9591	46.22	184.25	138.03	123.69	.09532	.24753	.34285
99	82.14	.012002	.9425	46.56	184.18	137.62	123.29	.09557	.24635	.34229
100	84.52	.012037	.9262	46.90	184.10	137.20	122.89	.09657	.24516	.34173
105	91.85	.012110	.8498	48.88	183.72	135.14	120.88	.09958	.23934	.33892
110	99.76	.012190	.7804	50.26	183.31	133.05	118.86	.10254	.23357	.33611
115	108.02	.012275	.7174	51.93	182.85	130.92	116.82	.10546	.22783	.33329
120	120.93	.012360	.6598	53.58	182.36	128.78	114.70	.10829	.22217	.33046
125	126.43	.012445	.6079	55.31	181.82	126.51	112.67	.11120	.21639	.32759
130	136.48	.012530	.5595	56.85	181.24	124.39	110.57	.11376	.21096	.32472
135	147.21	.012620	.5158	58.47	180.62	122.15	108.45	.11640	.20542	.32182
140	158.61	.012720	.4758	60.04	179.94	119.90	106.30	.11893	.19990	.31888

TABLE 56*a*

Saturated Sulphur Dioxide: Pressure Table

Pressure, Pound-inch^2 Absolute p	Temperature, Degrees F. t	Volume of Vapor, Foot^3-pound v''	Heat, B.t.u. per Pound: Liquid q	Heat, B.t.u. per Pound: Vapor i''	Entropy: Liquid s'	Entropy: Vapor s''
5	−25.40	14.47	4.32	180.45	.00988	.41552
10	− 1.34	7.520	12.00	182.95	.02698	.40000
15	14.43	5.110	17.30	184.17	.03839	.39091
20	26.44	3.878	21.41	184.86	.04702	.38329
25	36.33	3.123	24.83	185.26	.05407	.37754
30	44.76	2.614	27.78	185.48	.06003	.37269
35	52.20	2.247	30.38	185.58	.06522	.36848
40	58.83	1.970	32.73	185.60	.06982	.36470
45	64.85	1.754	34.79	185.54	.07390	.36133
50	70.40	1.577	36.72	185.45	.07763	.35826
55	75.53	1.434	38.51	185.32	.08105	.35539
60	80.29	1.314	40.15	185.16	.08418	.35373
65	84.76	1.211	41.69	184.98	.08705	.35023
70	88.97	1.125	43.14	184.77	.08975	.34789
75	93.00	1.047	44.50	184.55	.09228	.34568
80	96.88	.9809	45.78	184.33	.09464	.34557

TABLE 56*b*

Properties of Superheated Sulphur Dioxide

Absolute Pressure in Pounds per Square inch

Temperature, Degrees F.	4 — *−32.60 Degrees*: v	i	s
Saturation	*17.30*	*179.57*	*.42043*
−20	18.40	181.5	.42487
−10	18.83	183.0	.42836
0	19.27	184.6	.43179
10	19.70	186.1	.43516
20	20.14	187.7	.43847
30	20.57	189.3	.44161
40	21.00	190.9	.44491
50	21.42	192.5	.44806
60	21.85	194.1	.45116
70	22.27	196.7	.45421
80	22.70	197.3	.45722
90	23.12	198.9	.46018
100	23.54	200.5	.46311
110	23.96	202.1	.46600
120	24.39	203.8	.46885
130	24.81	205.2	.47167
140	25.23	207.1	.47445
150	25.65	208.8	.47720
160	26.08	210.4	.47991
170	26.50	212.1	.48259
180	26.92	213.8	.48523

Temperature, Degrees F.	6 — *−19.37 Degrees*: v	i	s
Saturation	*12.22*	*181.14*	*.41151*
0	12.75	184.3	.41850
10	13.04	185.9	.42198
20	13.34	187.5	.42538
30	13.63	189.1	.42869
40	13.93	190.7	.43196
50	14.23	192.3	.43517
60	14.52	193.9	.43833
70	14.71	195.6	.44140
80	15.11	197.2	.44443
90	15.40	199.9	.44741
100	15.69	200.5	.45035
110	16.97	202.2	.45326
120	16.26	203.8	.45613
130	16.54	205.3	.45890
140	16.82	207.1	.46176
150	17.09	208.8	.46451
160	17.35	210.4	.46722
170	17.62	212.1	.46990
180	17.88	213.7	.47254
190	18.13	215.4	.47514
200	18.38	217.0	.47769

Temperature, Degrees F.	8 — *−8.90 Degrees*: v	i	s
Saturation	*9.220*	*182.23*	*.40482*
0	9.516	183.7	.40871
10	9.751	185.4	.41230
20	9.983	187.1	.41579
30	10.21	188.8	.41922
40	10.44	190.5	.42256
50	10.66	192.2	.42582
60	10.88	193.8	.42903
70	11.10	195.5	.43216
80	11.32	197.1	.43524
90	11.54	198.8	.43825
100	11.75	200.4	.44123
110	11.97	202.1	.44416
120	12.18	203.7	.44705
130	12.39	205.4	.44990
140	12.61	207.0	.45271
150	12.82	208.8	.45543
160	13.03	210.3	.45820
170	13.24	212.0	.46089
180	13.46	213.6	.46353
190	13.66	215.3	.46614
200	13.88	216.9	.46871

TABLE 56*b*—*Continued*

Temperature, Degrees F.	10 *−1.34 Degrees*		
	v	*i*	*s*
Saturation	*7.520*	*182.95*	*.40000*
0	7.545	183.2	.40046
10	7.744	185.0	.40432
20	7.939	186.7	.40802
30	8.030	188.4	.41159
40	8.316	190.1	.41505
50	8.500	191.8	.41837
60	8.681	193.5	.42161
70	8.860	195.2	.42480
80	9.038	196.9	.42795
90	9.214	198.6	.43104
100	9.389	200.3	.43407
110	9.563	202.0	.43705
120	9.736	203.7	.43997
130	9.908	205.4	.44283
140	10.08	207.1	.44565
150	10.25	208.8	.44842
160	10.42	210.5	.45116
170	10.59	212.2	.45296
180	10.76	213.8	.45651
190	10.93	215.4	.45913
200	11.10	217.0	.46171

Temperature, Degrees F.	15 *14.43 Degrees*		
	v	*i*	*s*
Saturation	*5.110*	*184.17*	*.39091*
20	5.192	185.4	.39270
30	5.333	187.3	.39672
40	5.470	189.2	.40054
50	5.604	191.0	.40424
60	5.734	192.8	.40777
70	5.862	195.6	.41116
80	5.988	196.4	.41443
90	6.112	198.2	.41765
100	6.233	199.9	.42076
110	6.353	201.6	.42383
120	6.471	203.3	.42682
130	6.588	205.6	.48976
140	6.705	206.7	.43264
150	6.821	208.4	.43548
160	6.937	210.1	.43825
170	7.052	211.8	.44097
180	7.167	213.5	.44366
190	7.282	215.2	.44630
200	7.396	216.9	.44889

Temperature, Degrees F.	20 *26.44 Degrees*		
	v	*i*	*s*
Saturation	*3.878*	*184.86*	*.38329*
40	4.035	187.8	.38959
50	4.145	189.8	.39346
60	4.251	191.8	.39719
70	4.354	193.7	.40080
80	4.451	195.6	.40429
90	4.552	197.5	.40758
100	4.648	199.3	.41093
110	4.742	201.1	.41415
120	4.834	202.9	.41726
130	4.925	204.7	.42027
140	5.015	206.5	.42322
150	5.104	208.2	.42613
160	5.193	209.9	.42898
170	5.281	211.6	.43176
180	5.369	213.3	.43449
190	5.456	215.0	.43716
200	5.542	216.7	.43977
210	5.629	218.4	.44234
220	5.715	220.1	.44488

Temperature, Degrees F.	25 *36.33 Degrees*		
	v	*i*	*s*
Saturation	*3.123*	*185.26*	*.37754*
40	3.181	186.1	.37927
50	3.273	188.4	.38372
60	3.363	190.6	.38795
70	3.451	192.7	.39198
80	3.536	194.7	.39582
90	3.618	196.7	.39945
100	3.696	198.6	.40291
110	3.772	200.5	.40625
120	3.848	202.4	.40949
130	3.923	204.2	.41261
140	3.998	206.0	.41568
150	4.073	207.8	.41866
160	4.145	209.6	.42156
170	4.216	211.4	.42439
180	4.287	213.2	.43717
190	4.358	215.0	.42988
200	4.428	216.7	.43253
210	4.498	218.4	.43413
220	4.567	220.1	.43769
230	4.637	221.8	.44023
240	4.706	223.5	.44275

Temperature, Degrees F.	30 *44.76 Degrees*		
	v	*i*	*s*
Saturation	*2.614*	*185.48*	*.37269*
60	2.747	189.3	.37969
70	2.830	191.6	.38428
80	2.907	193.8	.38848
90	2.980	195.9	.39236
100	3.052	197.9	.39603
110	3.122	199.9	.39955
120	3.189	201.8	.40293
130	3.254	203.7	.40619
140	3.318	205.6	.40935
150	3.381	207.5	.41241
160	3.443	209.3	.41539
170	3.504	211.1	.41829
180	3.565	212.9	.42112
190	3.625	214.7	.42387
200	3.685	216.5	.42657
210	3.744	218.3	.42921
220	3.803	220.1	.43180
230	3.861	221.9	.43438
240	3.919	223.6	.43691
250	3.977	225.3	.43942
260	4.035	227.0	.44188

Temperature, Degrees F.	40 *58.83 Degrees*		
	v	*i*	*s*
Saturation	*1.970*	*185.60*	*.36470*
60	1.980	185.9	.3654
70	2.064	188.7	.37064
80	2.121	191.3	.37544
90	2.185	193.6	.37992
100	2.246	196.1	.38415
110	2.304	198.3	.38810
120	2.360	200.4	.39183
130	2.413	202.5	.39541
140	2.465	204.6	.39881
150	2.515	206.5	.40209
160	2.565	208.5	.41525
170	2.614	210.4	.41831
180	2.662	212.3	.41127
190	2.709	214.2	.41416
200	2.755	216.0	.41694
210	2.800	217.9	.41966
220	2.845	219.7	.42233
230	2.889	221.5	.42494
240	2.933	223.3	.42751
250	2.977	225.1	.43007
260	3.021	227.0	.43262

TABLE 56b—*Continued*

Temperature, Degrees F.	50 *70.40 Degrees* v	i	s
Saturation	*1.577*	*185.45*	*.35826*
80	1.668	188.4	.36366
90	1.723	191.2	.36887
100	1.775	193.9	.37369
110	1.825	196.4	.37815
120	1.872	198.8	.38234
130	1.917	201.1	.38627
140	1.961	203.3	.38998
150	2.003	205.4	.39353
160	2.044	207.5	.39691
170	2.084	209.6	.40015
180	2.123	211.6	.40327
190	2.161	213.4	.40628
200	2.199	215.4	.40919
210	2.237	217.3	.41200
220	2.274	219.2	.41477
230	2.311	221.1	.41748
240	2.347	223.0	.42015
250	2.383	224.9	.42275
260	2.418	226.7	.42535
270	2.451	228.5	.42791
280	2.489	230.3	.43045

Temperature, Degrees F.	60 *80.29 Degrees* v	i	s
Saturation	*1.3144*	*185.16*	*.35272*
100	1.288	191.4	.36403
110	1.346	194.3	.36906
120	1.403	197.0	.37375
130	1.459	199.5	.37810
140	1.514	201.9	.38217
150	1.563	204.2	.38603
160	1.608	206.5	.38963
170	1.650	208.6	.39310
180	1.689	210.7	.39639
190	1.726	212.8	.39956
200	1.751	214.8	.40260
210	1.785	216.8	.40554
220	1.819	218.7	.40839
230	1.853	220.7	.41118
240	1.885	222.6	.41391
250	1.917	224.5	.41657
260	1.948	226.4	.41917
270	1.979	228.2	.42175
280	2.010	230.1	.42431
290	2.040	232.0	.42685
300	2.070	233.8	.42935

Temperature, Degrees F.	70 *88.97 Degrees* v	i	s
Saturation	*1.125*	*184.77*	*.34789*
100	1.181	187.6	.35443
110	1.228	191.6	.36020
120	1.272	194.8	.36545
130	1.313	197.6	.37028
140	1.352	200.3	.37478
150	1.389	202.9	.37897
160	1.424	205.3	.38291
170	1.457	207.6	.38662
180	1.489	209.9	.39014
190	1.521	212.0	.39348
200	1.551	214.1	.39670
210	1.580	216.1	.39978
220	1.608	218.1	.40275
230	1.636	200.1	.40564
240	1.664	222.1	.40845
250	1.691	224.1	.41120
260	1.718	226.0	.41389
270	1.745	227.9	.41653
280	1.771	229.8	.41912
290	1.798	231.7	.42167
300	1.824	233.5	.42418

Temperature, Degrees F.	80 *96.88 Degrees* v	i	s
Saturation	*.9809*	*184.33*	*.34357*
100	.993	185.6	.34571
110	1.040	189.1	.35214
120	1.084	192.5	.35797
130	1.125	195.7	.36330
140	1.163	198.6	.36819
150	1.199	201.3	.37270
160	1.232	203.9	.37692
170	1.263	206.4	.38093
180	1.292	208.7	.38461
190	1.370	211.0	.38813
200	1.347	213.3	.39150
210	1.374	215.5	.39471
220	1.400	217.5	.39780
230	1.426	219.6	.40079
240	1.451	221.6	.40369
250	1.476	223.6	.40651
260	1.500	225.6	.40926
270	1.524	227.6	.41195
280	1.547	229.5	.41459
290	1.570	231.5	.41719
300	1.593	233.4	.41974

Temperature, Degrees F.	100 *110.15 Degrees* v	i	s
Saturation	*.7786*	*183.30*	*.33603*
120	.8190	187.3	.34296
130	.8575	191.0	.34942
140	.8928	194.6	.35528
150	.9255	197.9	.36061
160	.9561	200.9	.36558
170	.9848	203.7	.37009
180	1.012	206.4	.37431
190	1.038	209.0	.37829
200	1.062	211.5	.38203
210	1.086	213.8	.38556
220	1.109	216.1	.38892
230	1.131	218.4	.39214
240	1.152	220.5	.39524
250	1.173	222.6	.39824
260	1.194	224.7	.40114
270	1.213	226.8	.40397
280	1.232	228.8	.40673
290	1.251	230.8	.40944
300	1.268	232.8	.41207
310	1.284	234.8	.41464
320	1.299	236.7	.41716

Temperature, Degrees F.	120 *121.52 Degrees* v	i	s
Saturation	*.6430*	*182.19*	*.33954*
140	.7085	190.1	.34264
150	.7403	193.9	.34904
160	.7700	197.4	.35484
170	.7972	200.6	.36012
180	.8228	203.7	.36494
190	.8470	206.7	.36936
200	.8699	209.4	.37348
210	.8916	212.0	.37737
220	.9124	214.5	.38104
230	.9324	217.0	38451
240	.9515	219.3	.38785
250	.9700	221.5	.39106
260	.9880	223.7	.39416
270	1.006	225.9	.39713
280	1.023	228.0	.40002
290	1.040	230.1	.40284
300	1.056	233.2	.40558
310	1.072	234.3	.40825
320	1.088	236.3	.41085
330	1.104	238.3	.41338
340	1.120	240.3	.41583

Temperature, Degrees F.	140 *131.64 Degrees* v	i	s
Saturation	*.5451*	*181.04*	*.32388*
140	.5734	185.1	.33089
150	.6055	189.7	.33777
160	.6345	193.6	.34442
170	.6613	196.3	.35041
180	.6861	200.8	.35588
190	.7092	204.0	.36088
200	.7309	207.1	.36548

Temperature, Degrees F.	140 *131.64 Degrees* v	i	s
Saturation	*.5451*	*181.04*	*.32388*
210	.7513	210.0	.36976
220	.7707	212.7	.37379
230	.7892	215.4	.37758
240	.8070	217.9	.38118
250	.8241	220.3	.38461
260	.8405	222.6	.38789
270	.8564	224.9	.39105

Temperature, Degrees F.	140 *131.64 Degrees* v	i	s
Saturation	*.5451*	*181.04*	*.32388*
280	.8720	227.1	.39408
290	.8970	229.3	.39701
300	.9017	231.5	.39985
310	.9161	233.6	.40261
320	.9302	235.7	.40529
330	.9441	237.7	.40791
340	.9579	239.7	.41049

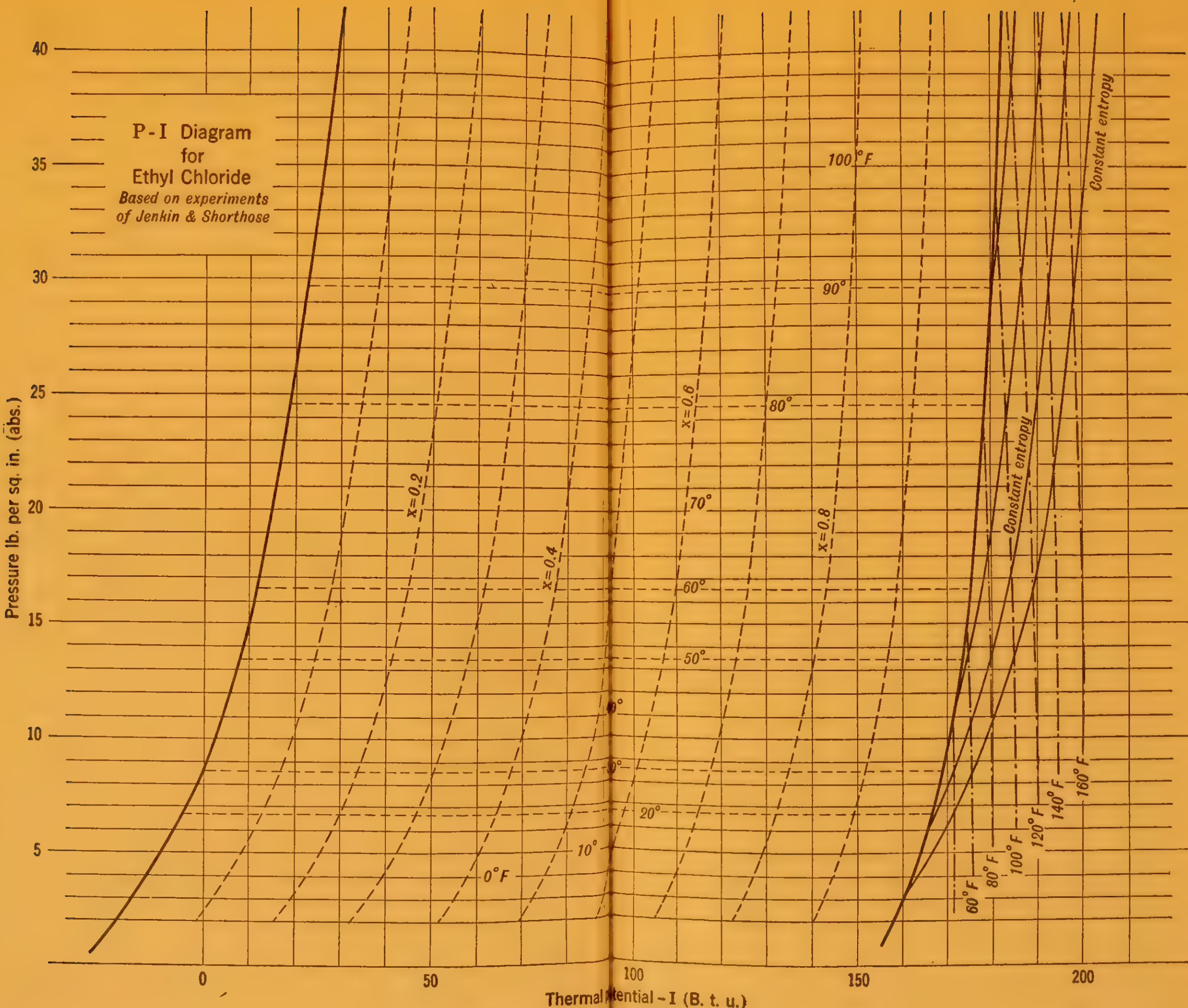

FIG. 167.—P–I Diagram for Ethyl Chloride.

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TABLE 57

Properties of Ethyl Chloride

Temperature, Degrees F.	Pressure, Pounds per Square Inch, Absolute	Weight, 1 Cubic Foot of Liquid	Specific Volume of Saturated Gas, Cubic Feet	Thermal Potential of the Liquid, B.t.u.	Latent Heat of Vaporization, B.t.u.	Thermal Potential of Saturated Vapor, B.t.u.
−24	1.90	60.40	32.85	−20.4	178.2	157.8
−22	2.02	60.30	31.13	−19.6	177.9	158.3
−20	2.16	60.20	29.54	−19.0	177.6	158.6
−18	2.31	60.10	28.12	−18.2	177.2	159.0
−16	2.45	60.01	26.73	−17.5	176.9	159.4
−14	2.61	59.93	25.41	−16.8	176.6	159.8
−12	2.77	59.85	24.14	−16.1	176.3	160.2
−10	2.94	59.78	22.95	−15.4	175.9	160.5
− 8	3.13	59.70	21.84	−14.6	175.6	161.0
− 6	3.31	59.60	20.76	−13.9	175.3	161.4
− 4	3.51	59.50	19.80	−13.1	174.9	161.8
− 2	3.71	59.40	18.89	−12.4	174.6	162.2
− 0	3.93	59.30	18.04	−11.7	174.2	162.5
+ 2	4.15	59.20	17.22	−11.0	173.9	162.9
+ 4	4.38	59.10	16.46	−10.2	173.6	163.4
6	4.61	58.99	15.74	− 9.5	173.2	163.7
8	4.86	58.90	15.03	− 8.8	172.9	164.1
10	5.13	58.79	14.36	− 8.1	172.6	164.5
12	5.42	58.69	13.74	− 7.3	172.2	164.9
14	5.71	58.59	13.14	− 6.6	171.9	165.3
16	6.01	58.48	12.58	− 5.9	171.6	165.7
18	6.32	58.39	12.06	− 5.1	171.3	166.2
20	6.65	58.29	11.56	− 4.4	170.9	166.5
22	6.98	58.20	11.07	− 3.6	170.6	167.0
24	7.31	58.10	10.58	− 2.9	170.2	167.3
26	7.65	58.00	10.11	− 2.1	169.9	167.8
28	8.01	57.89	9.66	− 1.4	169.6	168.2
30	8.40	57.78	9.22	− 0.6	169.2	168.6
32	8.80	57.66	8.81	+ 0.1	168.9	169.0
34	9.21	57.55	8.44	0.9	168.5	169.4
36	9.63	57.45	8.10	1.6	168.2	169.8
38	10.06	57.34	7.76	2.4	167.8	170.2
40	10.53	57.24	7.43	3.2	167.5	170.7
42	11.01	57.14	7.12	3.9	167.2	171.1
44	11.52	57.04	6.84	4.7	166.8	171.5
46	12.02	56.93	6.56	5.5	166.4	171.9
48	12.60	56.83	6.30	6.2	166.0	172.2
50	13.20	56.72	6.04	6.9	165.6	172.5
52	13.81	56.61	5.79	7.7	165.2	172.9
54	14.44	56.51	5.54	8.5	164.8	173.3
56	15.11	56.40	5.30	9.3	164.4	173.7
58	15.78	56.29	5.07	10.0	164.0	174.0
60	16.45	56.19	4.84	10.8	163.6	174.4
62	17.14	56.10	4.64	11.6	163.2	174.8
64	17.82	56.00	4.46	12.4	162.8	175.2
66	18.52	55.91	4.30	13.1	162.4	175.5
68	19.25	55.80	4.15	13.9	162.0	175.9
70	21.03	55.69	4.00	14.7	161.6	176.3
72	21.83	55.58	3.86	15.4	161.2	176.8
74	21.66	55.48	3.70	16.2	160.8	177.0
76	22.55	55.38	3.60	17.0	160.4	177.4
78	23.42	55.26	3.49	17.8	160.0	177.8
80	24.33	55.15	3.39	18.6	159.5	178.1
82	25.25	55.04	3.30	19.4	159.1	178.5
84	26.19	54.94	3.21	20.2	158.7	178.9
86	27.15	54.83	3.13	21.0	158.2	179.2
88	28.17	54.72	3.06	21.7	157.8	179.5
90	29.24	54.61	2.99	22.6	157.3	179.9
92	30.33	54.50	2.92	23.4	156.8	180.2
94	31.45	54.40	2.86	24.1	156.3	180.4
96	32.60	54.30	2.80	25.0	155.8	180.8
98	33.75	54.20	2.75	25.8	155.4	181.2
100	34.93	54.10	2.70	26.6	154.9	181.5
102	36.14	54.00	2.66	27.4	154.5	181.9
104	37.36	53.90	2.63	28.2	154.0	182.2
106	38.59	53.79	2.58	29.0	153.6	182.6
108	39.82	53.68	2.54	29.8	153.1	182.9
110	41.10	53.56	2.51	30.5	152.6	183.1

TABLE 58

THERMODYNAMIC PROPERTIES OF METHYL CHLORIDE (CH_3CL)

	Temperature, Degrees F.	Pressure (Absolute)	Pressure (Gage)	Specific Volume of Saturated Gas, Cubic Feet per Pound	Density of Liquid, Pounds per Cubic Foot	Total Heat, Liquid, B.t.u. per Pound	Latent Heat, B.t.u. per Pound	Heat of Vapor, B.t.u. per Pound
	−24	10.60	8.4*	8.83	63.435	−20.45	187.22	166.77
†	−22	11.20	7.4*	8.466	63.310	−19.68	186.80	167.12
	−20	11.75	6.1*	8.09	63.185	−19.0	186.36	167.36
	−18	12.50	4.8*	7.73	63.060	−18.25	185.92	167.67
	−16	13.1	3.2*	7.38	62.935	−17.57	185.50	167.93
	−14	13.75	1.4*	7.06	62.180	−16.81	185.07	168.26
	−12	14.4	0.8*	6.75	62.685	−16.10	184.64	168.54
	−10	15.0	0.3	6.46	62.560	−15.38	184.21	168.83
	− 8	15.8	1.1	6.18	62.435	−14.68	183.78	169.10
	− 6	16.5	1.8	5.92	62.310	−13.95	183.33	169.38
†	− 4	17.08	2.3	5.67	62.185	−13.20	182.90	169.70
	− 2	17.9	3.2	5.41	62.061	−12.50	182.44	169.94
	0	18.8	4.1	5.18	61.936	−11.75	181.98	170.23
	2	19.6	4.9	4.96	61.811	−11.0	181.52	170.52
	4	20.5	5.8	4.75	61.686	−10.30	181.08	170.78
	6	21.5	6.8	4.55	61.561	− 9.55	180.60	171.05
	8	22.4	7.7	4.36	61.436	− 8.80	180.12	171.32
	10	23.3	8.6	4.18	61.311	− 8.06	179.65	171.59
	12	24.4	9.7	4.02	61.187	− 7.30	179.18	171.88
†	14	25.34	10.6	3.86	61.086	− 6.62	178.70	172.08
	16	26.5	11.8	3.70	60.959	− 5.85	178.23	172.40
	18	27.6	12.9	3.55	60.831	− 5.10	177.75	172.65
	20	28.8	14.1	3.41	60.702	− 4.32	177.27	172.95
	22	29.8	15.1	3.28	60.593	− 3.60	176.80	173.20
	24	31.2	16.5	3.15	60.464	− 2.85	176.34	173.49
	26	32.5	17.8	3.03	60.365	− 2.10	175.87	173.77
	28	33.8	19.1	2.92	60.206	− 1.38	175.38	174.00
	30	35.2	20.5	2.81	60.077	− 0.62	174.90	174.28
†	32	36.57	21.8	2.69	59.914	+ 0.11	174.40	174.51
	34	37.9	23.2	2.59	59.779	0.87	173.92	174.79
	36	39.5	24.8	2.49	59.650	1.63	173.41	175.04
	38	41.1	26.4	2.40	59.521	2.40	172.91	175.31
	40	42.6	27.9	2.31	59.492	3.15	172.42	175.57
	42	44.3	29.6	2.225	59.263	3.90	171.92	175.82
	44	46.1	31.4	2.14	59.134	4.66	171.42	176.08

	46	47.8	33.1	2.06	59.005	5.41	170.90	176.31
	48	49.6	34.9	1.99	58.876	6.19	170.40	176.59
†	50	51.51	36.77	1.93	58.747	6.88	169.90	176.78
	52	53.5	38.8	1.86	58.616	7.66	169.40	177.06
	54	55.4	40.7	1.79	58.484	8.40	168.90	177.30
	56	57.4	42.7	1.72	58.353	9.20	168.40	177.60
	58	59.3	44.6	1.67	58.220	9.95	167.88	177.83
	60	61.6	46.9	1.61	58.077	10.70	167.35	178.05
	62	63.8	49.1	1.55	57.943	11.48	166.83	178.31
	64	66.1	51.4	1.50	57.809	12.25	166.29	178.54
	66	68.4	53.7	1.45	57.675	13.00	165.74	178.74
†	68	70.97	56.23	1.39	57.541	13.75	165.20	178.95
	70	73.3	58.6	1.34	57.403	14.52	164.65	179.17
	72	75.8	61.1	1.30	57.265	15.30	164.10	179.40
	74	78.5	63.8	1.26	57.127	16.08	163.56	179.64
	76	79.8	65.1	1.22	56.989	16.83	163.00	179.83
	78	82.5	67.8	1.17	56.851	17.59	162.44	180.03
	80	85.3	70.6	1.14	56.714	18.36	161.88	180.24
	82	89.7	75.0	1.10	56.576	19.12	161.31	180.43
	84	92.7	78.0	1.06	56.438	19.88	160.75	180.63
†	86	95.52	80.78	1.04	56.300	20.64	160.20	180.84
	88	98.9	84.2	1.00	56.161	21.34	159.65	180.99
	90	102.1	87.4	0.98	56.022	22.13	159.09	181.22
	92	105.3	90.6	.95	55.883	22.90	158.52	181.42
	94	108.6	93.9	.92	55.744	23.68	157.98	181.66
	96	112.0	97.3	.892	55.605	24.46	157.41	181.87
	98	115.3	100.6	.88	55.466	25.27	156.86	182.13
	100	118.8	104.1	.85	55.327	26.06	156.30	182.36
	102	122.5	107.8	.83	55.188	26.84	155.75	182.59
†	104	126.2	111.5	.818	55.050	27.63	155.20	182.83
	106	130.0	115.3	.79	54.911	28.41	154.62	183.02
	108	133.9	119.2	.78	54.772	29.22	154.04	183.26
	110	137.6	122.9	.765	54.633	30.03	153.46	183.49
	112	141.9	127.2	.745	54.494	30.84	152.90	183.74
	114	146.1	131.4	.730	54.355	31.67	152.33	184.00
	116	150.5	135.8	.722	54.216	32.50	151.76	184.26
	118	155.0	140.3	.712	54.076	33.33	151.19	184.52
	120	159.6	144.9	.700	53.936	34.16	150.62	184.78
	122	164.4	149.7	.699	53.798	34.99	150.05	185.04
	124	169.2	154.5	.689	53.664	35.82	149.58	185.40
	126	174.3	159.6	.678	53.530	36.65	149.01	185.66
	128	179.2	164.5	.668	53.496	37.48	148.44	185.92
	130	184.3	169.6	.659	53.362	38.31	147.87	186.18

* Inches of Mercury

† Values indicated by † are from D. N. Shorthose. Other values by H. J. Macintire.

in many cases in which the required piston displacement can be secured with less difficulty than in the case of the reciprocating compressor.

Methyl Chloride.—Methyl chloride (CH_3Cl) has considerable resemblance to ethyl chloride, but the unit pressures are greater and the piston displacement per ton of refrigeration is less. It is slightly anæsthetic but not unpleasant to inhale except that some persons are nauseated by the gas, and it is inflammable; much less so than ethyl chloride, but more so than ammonia. The latent heat of vaporization (r) is about 170 B.t.u. per lb. It does not seem to be corrosive to copper or compositions of copper, nor to iron or steel, the pressures encountered are 95 lb. per sq. in. abs. (80 lb. gage) or less as a rule (Table 58), and the evaporating pressures are greater than one atmosphere if the boiling temperature is higher than -10 deg. F.

Methyl chloride[5] does not appear to cause any damage to goods or commodities. Both methyl and ethyl chloride are nearly as harmless as carbon dioxide, as a relatively large percentage of these gases in the air will not injure the eyes or lungs and seemingly the bad effects are only from the absence of oxygen, as would be the case with carbon dioxide or nitrogen. There does not seem to be any *cumulative* effect, as would be the case with *carbon monoxide*, which, incidentally, may not show the real effect on a person until some time after the exposure. Both methyl and ethyl chloride must be anhydrous, or the water content will freeze at the expansion valve. Mineral oils dissolve readily in methyl and ethyl chloride, but this is not true in the case of sulphur dioxide.

Propane.—Of the gasoline compounds, butane (C_4H_{10}), propane (C_3H_8) and ethane (C_2H_6), propane approximates very closely the physical characteristics—as regards pressures and refrigerating ability—of ammonia, whereas ethane is more like carbon dioxide and butane like sulphur dioxide or ethyl chloride. Propane will give a lower refrigerating temperature than ammonia will for the same suction pressure. According to Dr. E. E. Smith, propane up to 50 per cent in the air (by volume) will not cause injury to people. Table 60[6] gives the limits of the explosibility of propane and other refrigerants. Table 59 gives some of the thermodynamic properties of propane.

Butane.—As mentioned, butane has the pressures found in sulphur dioxide practice but it is harmless to breathe. Its particular field—up to the present time—is in the household or the $\frac{1}{6}$-ton (ice cream cabinet) machine now being manufactured in considerable numbers. In either

[5] Certain Physical and Chemical Properties of Methyl Chloride, by H. J. Macintire, C. S. Marvel and S. F. Ford, Amer. Soc. of Refrigerating Engineers, 1927.

[6] H. D. Edwards, Amer. Soc. Refrigerating Engineers, 1924.

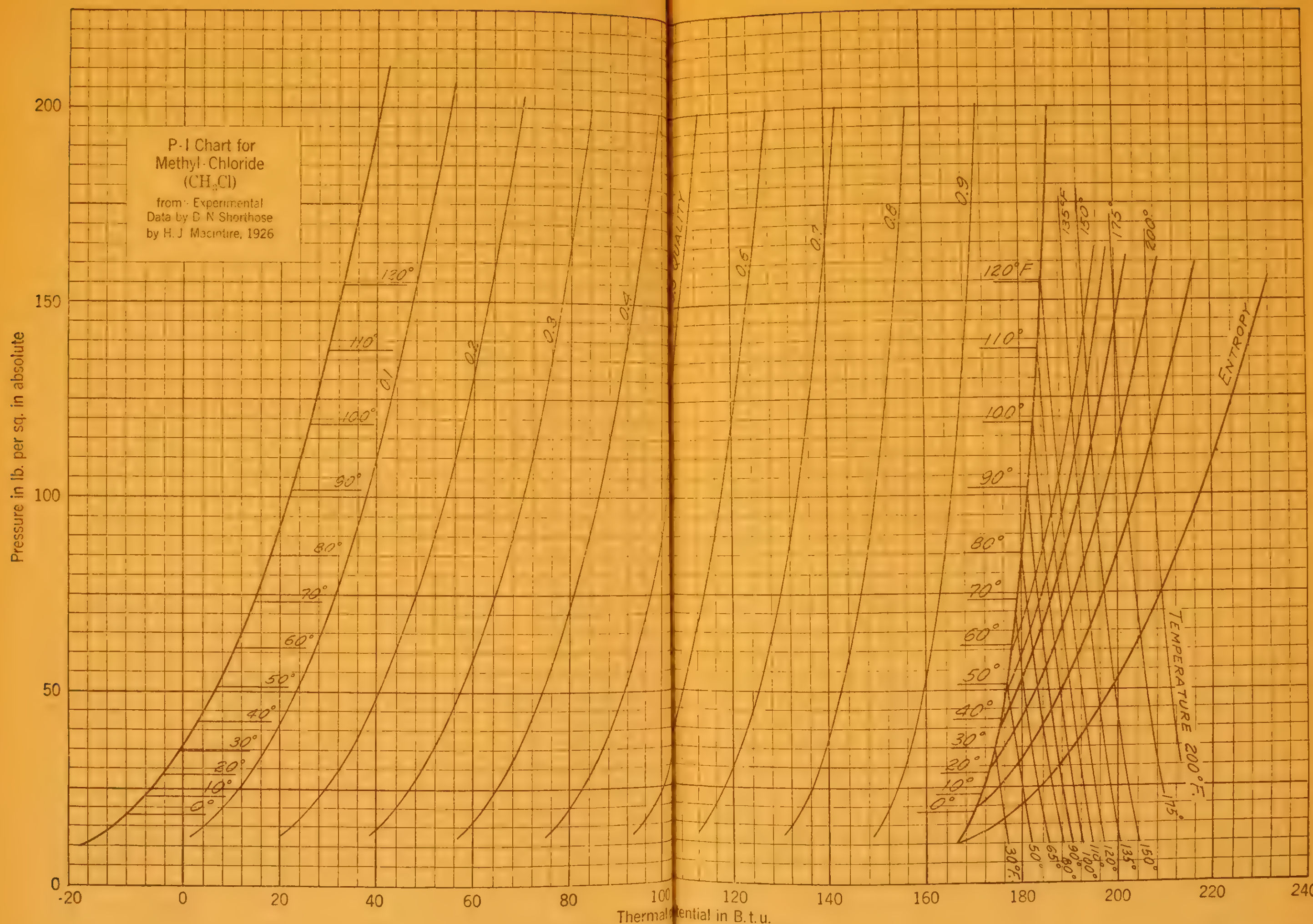

Fig. 168.—P-I Diagram for Methyl Chloride.

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TABLE 59
Thermodynamic Properties of Propane

Temperature, Degrees F.	Pressure, Pounds per Square Inch		Specific Volume, Cubic Feet per Pound		Density, Pounds per Cubic Foot		Heat Content, B.t.u. per Pound		Latent Heat, B.t.u. per Pound	Entropy, B.t.u. per Pound, Deg. F.		Temperature, Degrees F.
t	Absolute p	Gage $g.p.$	Liquid v	Vapor V	Liquid $1/v$	Vapor $1/V$	Liquid h	Vapor H	L	Liquid s	Vapor S	t
−75	6.37	17.0*	0.02660	14.5	37.59	0.0690	−39.5	151.0	190.5	−0.092	0.404	−75
−70	7.37	14.9*	0.02674	12.9	37.40	0.0775	−37.0	152.5	189.5	−0.806	0.400	−70
−65	8.48	12.7*	0.02688	11.3	37.20	0.0885	−34.5	153.5	188.0	−0.080	0.397	−65
−60	9.72	10.1*	0.02703	9.93	37.00	0.111	−32.0	155.0	187.0	−0.074	0.393	−60
−55	11.1	7.3*	0.02717	8.70	36.80	0.115	−29.0	156.5	185.5	−0.067	0.391	−55
−50	12.6	4.3*	0.02732	7.74	36.60	0.129	−26.5	158.0	184.5	−0.061	0.389	−50
−45	14.4	0.6*	0.02748	6.89	36.39	0.145	−24.0	159.0	183.0	−0.055	0.386	−45
−40	16.2	1.5	0.02763	6.13	36.19	0.163	−21.5	160.0	181.5	−0.049	0.384	−40
−35	18.1	3.4	0.02779	5.51	35.99	0.181	−19.0	161.0	180.0	−0.042	0.382	−35
−30	20.3	5.6	0.02795	4.93	35.78	0.203	−16.0	163.0	179.0	−0.036	0.380	−30
−25	22.7	8.0	0.02811	4.46	35.58	0.224	−13.5	164.0	177.5	−0.030	0.378	−25
−20	25.4	10.7	0.02827	4.00	35.37	0.250	−11.0	165.0	176.0	−0.024	0.377	−20
−15	28.3	13.6	0.02844	3.60	35.16	0.278	− 8.0	167.0	175.0	−0.018	0.375	−15
−10	31.4	16.7	0.02860	3.26	34.96	0.307	− 5.5	168.0	173.5	−0.012	0.374	−10
− 5	34.7	20.0	0.02878	2.97	34.75	0.337	− 2.5	169.5	172.0	−0.006	0.372	− 5
0	38.2	23.5	0.02895	2.71	34.54	0.369	0	170.5	170.5	0.000	0.371	0
+ 5	41.9	27.2	0.02913	2.48	34.33	0.403	+ 3.0	172.0	169.5	+0.006	0.370	+ 5
10	46.0	31.3	0.02931	2.27	34.12	0.441	5.5	173.5	168.0	0.012	0.370	10
15	50.6	35.9	0.02950	2.07	33.90	0.483	8.5	175.0	166.5	0.018	0.369	15
20	55.5	40.8	0.02970	1.90	33.67	0.526	11.0	176.0	165.0	0.024	0.368	20
25	60.9	46.2	0.02991	1.74	33.43	0.575	14.0	177.5	163.5	0.030	0.368	25
30	66.3	51.6	0.03012	1.60	33.20	0.625	17.0	179.0	162.0	0.035	0.366	30
35	72.0	57.3	0.03033	1.48	32.97	0.676	20.0	180.5	160.5	0.041	0.366	35
40	78.0	63.3	0.03055	1.37	32.73	0.730	23.0	182.0	159.0	0.047	0.366	40
45	84.6	69.9	0.03078	1.27	32.49	0.787	26.0	183.5	157.5	0.053	0.365	45
50	91.8	77.1	0.03102	1.18	32.24	0.847	29.0	185.0	156.0	0.059	0.365	50
55	99.3	84.6	0.03125	1.10	32.00	0.909	32.0	186.5	154.5	0.065	0.365	55
60	107.1	92.4	0.03150	1.01	31.75	0.990	35.0	188.0	153.0	0.070	0.364	60
65	115.4	100.7	0.03174	0.945	31.50	1.06	38.0	189.5	151.5	0.076	0.364	65
70	124.0	109.3	0.03201	0.883	31.24	1.13	41.0	190.5	149.5	0.082	0.364	70
75	133.2	118.5	0.03229	0.825	30.97	1.21	44.0	192.0	148.0	0.088	0.364	75
80	142.8	128.1	0.03257	0.770	30.70	1.30	47.5	193.5	146.0	0.093	0.364	80
85	153.1	138.4	0.03287	0.722	30.42	1.39	50.5	195.0	144.5	0.099	0.364	85
86	155.3	140.5	0.03292	0.717	30.38	1.40	51.0	195.0	144.0	0.100	0.364	86
90	164.0	149.0	0.03317	0.673	30.15	1.49	54.0	196.5	142.5	0.105	0.364	90
95	175.0	160.0	0.03348	0.632	29.87	1.58	57.0	197.5	140.5	0.111	0.364	95
100	187.0	172.0	0.04481	0.591	29.58	1.69	60.5	199.0	138.5	0.116	0.363	100
105	200.0	185.0	0.03416	0.553	29.27	1.81	63.5	200.0	136.5	0.122	0.364	105
100	187.0	172.0	0.03381	0.591	29.58	1.69	67.0	201.0	134.0	0.128	0.363	110
115	226.0	211.0	0.03493	0.488	28.63	2.05	70.5	202.0	131.5	0.134	0.363	115
120	240.0	225.0	0.03534	0.459	28.30	2.18	73.5	202.5	129.0	0.140	0.363	120
125	254.0	239.0	0.03575	0.432	27.97	2.31	77.0	203.5	126.5	0.145	0.361	125

* Inches of mercury below one standard atmosphere (29.92 in.).

case medium pressures and safety are the first requirements, and the actual (relative) size is secondary, as is the amount of power required to operate the compressor. Tables 61 and 62 give the thermodynamic properties of butane and isobutane.

TABLE 60

Refrigerants	Explosion Limits with Air Ratio by Volume		Maximum Explosion Pressures, Pounds per Square Inch		Time of Maximum Pressure in Seconds	
	Low	High	*a*	*b*	*a*	*b*
Ammonia	13.1*	26.8	54	52	0.252	0.175
Butane	1.65	5.7	102	102	0.027	0.027
Carbon dioxide						
Ethane	3.1	10.7	....	108	....	0.018
Ethyl chloride	4.3	14.0	98	....	0.049	
Gasoline	1.4	6.0	....	100	....	0.026
Illuminating gas	7.0	21.0	....	95	....	0.017
Methyl chloride	8.9	15.5	81	....	0.099	
Propane	2.4	8.4	100	104	0.023	0.020
Sulphur dioxide						

* Lowenstein says the lower value is 19.6. *a* refers to Linde Air Products Laboratories. *b* refers to the Underwriters Laboratory, Chicago.

Dicloethylene.—Dicloethylene ($C_2H_2Cl_2$) is being used[7] by only one company in a centrifugal type of rotary compressor. It is a colorless liquid, practically non-explosive and harmless to people and commodities. The unit pressure at the standard condensing temperature of 86 deg. F. is about one-half an atmosphere.

This refrigerant is especially good for centrifugal compression because the molecular weight is high and the unit pressures are very low—being below the atmosphere—although the ratio of the suction to the condensing pressure is nominal. The result is that with *six* stages the actual pressure increase—$6\frac{1}{2}$ lb.—is easily obtained. It will be noted that all pressures are less than that of the atmosphere, and the difficulty, if any, is to remove the air that enters the compressor through the shaft stuffing boxes without excessive loss of the refrigerant. Table 63 which was prepared by Willis Carrier, gives the design requirements with this refrigerant.

[7] W. H. Carrier, American Soc. Refrig. Engineers, June, 1924.

TABLE 61

Thermodynamic Properties of Butane (C_4H_{10})

Temperature, Degrees F.	Pressure, Pounds per Square Inch		Specific Volume, Cubic Feet per Pound		Density, Pounds per Cubic Foot		Heat Content, B.t.u. per Pound		Latent Heat, B.t.u. per Pound	Entropy, B.t.u. per Pound, Deg. F.		Temperature, Degrees F.
	Absolute	Gage	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor		Liquid	Vapor	
t	p	$g\,p.$	v	V	$1/v$	$1/V$	h	H	L	s	S	t
0	7.3	15.0*	0.02591	11.1	38.59	0.0901	0	170.5	170.5	0.000	0.371	0
5	8.2	13.2*	0.02603	9.98	38.41	0.100	2.5	172.0	169.5	0.006	0.370	5
10	9.2	11.1*	0.02615	8.95	38.24	0.112	5.5	174.0	168.5	0.011	0.370	10
15	10.4	8.8*	0.02627	8.05	38.07	0.124	8.0	176.0	168.0	0.017	0.371	15
20	11.6	6.3*	0.02639	7.23	37.89	0.138	10.5	177.5	167.0	0.022	0.370	20
25	13.0	3.6*	0.02651	6.55	37.72	0.153	13.0	179.0	166.0	0.028	0.371	25
30	14.4	0.6*	0.02664	5.90	37.54	0.169	16.0	181.5	165.5	0.033	0.371	30
35	16.0	1.3	0.02676	5.37	37.37	0.186	19.0	183.5	164.5	0.039	0.371	35
40	17.7	3.0	0.02689	4.88	37.19	0.205	21.5	185.0	163.5	0.044	0.371	40
45	19.6	4.9	0.02703	4.47	37.00	0.224	24.5	187.0	162.5	0.050	0.372	45
50	21.6	6.9	0.02716	4.07	36.82	0.246	27.0	188.5	161.5	0.056	0.373	50
55	23.8	9.1	0.02730	3.73	36.63	0.268	30.0	190.5	160.5	0.061	0.373	55
60	26.3	11.6	0.02743	3.40	36.45	0.294	33.0	192.5	159.5	0.067	0.374	60
65	28.9	14.2	0.02759	3.12	36.24	0.321	36.0	194.5	158.5	0.072	0.374	65
70	31.6	16.9	0.02773	2.88	36.06	0.347	38.5	196.0	157.5	0.078	0.375	70
75	34.5	19.8	0.02789	2.65	35.86	0.377	41.5	198.0	156.5	0.083	0.375	75
80	37.6	22.9	0.02805	2.46	35.65	0.407	44.5	199.5	155.0	0.089	0.376	80
85	40.9	26.2	0.02821	2.28	35.45	0.439	47.5	201.5	154.0	0.094	0.376	85
86	41.6	26.9	0.02825	2.24	35.40	0.446	48.5	202.0	153.5	0.095	0.376	86
90	44.5	29.8	0.02838	2.10	35.24	0.476	51.0	203.0	152.0	0.100	0.377	90
95	48.2	33.5	0.02854	1.96	35.04	0.510	54.0	205.0	151.0	0.105	0.377	95
100	52.2	37.5	0.02870	1.81	34.84	0.552	57.0	206.5	149.5	0.111	0.378	100
105	56.4	41.7	0.02889	1.70	34.62	0.588	60.5	208.5	148.0	0.117	0.380	105
110	60.8	46.1	0.02906	1.58	34.41	0.633	63.5	210.5	147.0	0.122	0.380	110
115	65.6	50.9	0.02925	1.48	34.19	0.676	66.5	212.0	145.5	0.128	0.381	115
120	70.8	56.1	0.02945	1.38	33.96	0.725	70.0	213.5	143.5	0.134	0.382	120
125	76.0	61.3	0.02966	1.30	33.72	0.769	73.5	215.5	142.0	0.139	0.382	125
130	81.4	66.7	0.02986	1.21	33.49	0.826	76.5	217.0	140.5	0.145	0.384	130
135	87.0	72.3	0.03009	1.14	33.23	0.877	80.0	219.0	139.0	0.151	0.385	135
140	92.6	77.9	0.03032	1.07	32.98	0.934	83.5	221.0	137.5	0.157	0.386	140

* Inches of mercury below one standard atmosphere (29.92 in.)

TABLE 62

THERMODYNAMIC PROPERTIES OF ISOBUTANE (C_4H_{10})

Temperature, Degrees F.	Pressure, Pounds per Square Inch		Specific Volume, Cubic Feet per Pound		Density, Pounds per Cubic Foot		Heat Content, B.t.u. per Pound		Latent Heat, B.t.u. per Pound	Entropy, B.t.u. per Pound, Deg. F.		Temperature, Degrees F.
t	Absolute p	Gage $g.p.$	Liquid v	Vapor V	Liquid $1/v$	Vapor $1/V$	Liquid h	Vapor H	L	Liquid s	Vapor S	t
−20	7.50	14.6*	0.02610	10.5	38.35	0.0952	−9.0	156.5	165.5	−0.020	0.356	−20
−15	8.30	13.0*	0.02620	9.90	38.15	0.101	−7.0	157.0	164.0	−0.015	0.354	−15
−10	9.28	11.0*	0.02635	8.91	37.95	0.112	−4.5	168.5	163.0	−0.010	0.353	−10
− 5	10.4	8.8*	0.02645	7.99	37.80	0.125	−2.5	159.5	162.0	−0.005	0.351	− 5
0	11.6	6.3*	0.02660	7.17	37.60	0.139	0	160.5	160.5	0.000	0.350	0
+ 5	13.1	3.3*	0.02675	6.41	37.40	0.156	+2.5	162.0	159.5	0.005	0.348	+ 5
10	14.6	0.2*	0.02690	5.75	37.20	0.174	5.0	163.5	158.5	0.011	0.348	10
15	16.3	1.6	0.02705	5.18	37.00	0.193	7.5	164.5	157.0	0.016	0.347	15
20	18.2	3.5	0.02715	4.68	36.80	0.214	10.0	166.0	156.0	0.021	0.346	20
25	20.2	5.5	0.02730	4.24	36.60	0.236	13.0	167.5	154.5	0.027	0.346	25
30	22.3	7.6	0.02745	3.86	36.40	0.259	15.5	169.0	153.5	0.032	0.346	30
35	24.6	9.9	0.02760	3.52	36.20	0.284	18.0	170.5	152.5	0.038	0.346	35
40	26.9	12.2	0.02780	3.22	36.00	0.311	21.0	172.0	151.0	0.044	0.346	40
45	29.5	14.8	0.02795	2.96	35.80	0.338	24.0	174.0	150.0	0.049	0.346	45
50	32.5	17.8	0.02810	2.71	35.60	0.369	27.0	175.5	148.5	0.055	0.346	50
55	35.5	20.8	0.02825	2.49	35.40	0.402	30.0	177.5	147.5	0.061	0.347	55
60	38.7	24.0	0.02840	2.28	35.20	0.439	33.0	179.0	146.0	0.067	0.348	60
65	42.2	27.5	0.02855	2.10	35.00	0.476	36.5	181.0	144.5	0.073	0.349	65
70	45.8	31.1	0.02875	1.94	34.80	0.515	39.5	183.0	143.5	0.079	0.350	70
75	49.7	35.0	0.02890	1.79	34.60	0.559	43.0	185.0	142.0	0.086	0.351	75
80	53.9	39.2	0.02910	1.66	34.35	0.602	46.5	187.0	140.5	0.092	0.352	80
85	58.6	43.9	0.02930	1.54	34.10	0.649	50.0	189.0	139.0	0.098	0.353	85
86	59.5	44.8	0.02935	1.52	34.10	0.658	50.5	189.5	139.0	0.099	0.354	86
90	63.3	48.6	0.02950	1.42	33.90	0.704	53.5	191.0	137.5	0.105	0.356	90
95	68.4	53.7	0.02965	1.32	33.70	0.758	57.5	193.5	136.0	0.112	0.358	95
100	73.7	59.0	0.02990	1.23	33.45	0.813	61.0	195.5	134.5	0.118	0.359	100
105	79.3	64.6	0.03005	1.14	33.25	0.877	65.0	198.0	133.0	0.125	0.360	105
110	85.1	70.4	0.03030	1.07	33.00	0.935	69.0	200.0	131.0	0.132	0.362	110
115	91.4	76.7	0.03050	0.990	32.80	1.01	73.0	202.5	129.5	0.139	0.364	115
120	98.0	83.3	0.03075	0.926	32.50	1.08	77.0	204.5	127.5	0.147	0.367	120
125	104.8	90.1	0.03095	0.867	32.30	1.15	81.5	207.5	126.0	0.154	0.369	125
130	112.0	97.3	0.03125	0.811	32.00	1.23	86.0	210.0	124.0	0.161	0.371	130
135	119.3	104.6	0.03145	0.760	31.80	1.32	90.5	212.5	122.0	0.169	0.375	135
140	126.8	112.1	0.03175	0.716	31.80	1.32	95.0	215.5	120.5	0.176	0.377	140

* Inches of mercury below one standard atmosphere (29.92 in.).

TABLE 63

On the Basis of 200 B.t.u. Refrigeration per Minute and 86 Degrees Condensing and 5 Degrees Evaporating Temperature

$C_2H_2Cl_2$	Weight of Refrigeration, Pounds	Volume in Cubic Feet per Minute	In Inch of Mercury		Ratio $\frac{p_2}{p_1}$	Coefficient of Performance	Horse Power per Ton
			p_1	p_2			
	1.768	108.4	1.78	14.65	8.23	5.14	0.918

TABLE 64

Properties of Saturated Ethane

Temperature, Degrees F.	Pressure, Pounds per Square Inch		Heat Content above 32 Deg. F., B.t.u. per Pound		Latent Heat, B.t.u. per Pound, Total
	Gage	Absolute	Liquid	Vapor	
−40.0	100.0	114.7	−32.4	149.6	182.0
−30.0	120.3	135.0	−27.9	150.4	178.4
−20.0	144.8	159.5	−23.4	150.6	174.0
−10.0	173.3	188.0	−18.9	150.3	169.2
0.0	204.3	219.0	−14.4	149.4	163.8
10.0	238.3	253.0	− 9.9	148.1	158.0
20.0	277.3	292.0	− 5.4	145.7	151.1
30.0	318.3	333.0	− 0.9	142.1	143.0
40.0	361.3	376.0	4.0	137.1	133.1
50.0	410.3	425.0	9.4	130.8	121.4
60.0	466.3	481.0	15.2	122.1	106.9
70.0	532.3	547.0	22.2	111.4	89.2
80.0	610.3	625.0	32.3	98.2	65.9
89.8	703.4	718.1	....	....	0.0

Mixtures of Different Refrigerants.—Various attempts have been made to mix different refrigerants with the idea that a new "compound" will be obtained or something that will give the characteristics in between the extremes of the two being mixed. This has been tried with carbon dioxide and sulphur dioxide, methyl and ethyl chloride, and butane and propane. Such attempts have always been failures, as a new compound is *never* obtained, and leaks occurring soon change the proportions of the mixture. In fact, if leaks do occur, it is the vapor with the higher tension that escapes in the largest proportion. As the mixture is not a pure

compound, and the proportion is not known accurately, pressures in the system cannot be interpreted with any accuracy.

Water Vapor.—Water vapor has been proposed for years, especially for air cooling where the refrigerating temperature needs to be between 45 and 60 deg. F. If steam is used as a refrigerant under these conditions, the evaporating pressures have to be extremely low, and the specific volume of the saturated vapor is very great. The only method seriously proposed for this condition is the Leblanc system of compressing with water pistons, similar to the method of securing high vacuum in steam condensers. Although a number of applications to warship and other refrigeration has been made in France, little has been attempted in the United States. The advantage would be perfect safety, the "brine" used being pure water, yet the horse power per ton is considerably higher than would be the case with other refrigerants.

Air.—Air as a refrigerant was one of the first successful means of securing mechanical refrigeration, and it has many good features. Air is cheap and plentiful and leaks are harmless except from the point of view of the loss of capacity or the cost in dehumidifying it. Yet it has been discontinued in all except a very few installations because of the bulk of the compressor, operating troubles, and the higher horse power per ton of refrigeration than would be necessary with the volatile liquids now used so generally. In order to secure low temperatures an expansion cylinder *must* be used which returns some of the power required for compression back to the shaft. In order to decrease the piston displacement per ton of refrigeration a *dense* air usually is used, of three or more atmospheres' suction pressure, in which case the cycle is a closed one. Although there are decided advantages in the use of air, yet it seems unlikely that one would face the operating troubles with the lubrication at the very low temperatures, from −50 to −100 deg. F., and with the continued trouble with the trace of water vapor which cannot be removed from the air and which will freeze in the valves of the expansion cylinder at any moment; a condition which occurred almost daily in the dense air machines in the U. S. navy. Its use in the future is very doubtful when it is considered that very much better results can be obtained by the use of the many available refrigerants of the volatile liquid type (see Chapter I).

Lubrication.—Lubrication of the compressor cylinder is one of the very important points to be considered. The difficulty in refrigeration is that the refrigerant frequently is absorbed by the oil or vice versa, or that some chemical action takes place. The lubricant must not freeze at the lowest temperature used, usually taken at −20 deg. F., and must not show a flash test at less than 300 deg. F. Experience has shown

that a straight, refined, mineral oil with a viscosity of about 130 to 180 Saybolt at 104 deg. F. is about correct for most work. The presence of water[8] in the ammonia or the oil will tend towards an emulsifying effect due to the ammonia hydrate formed.

In the case of carbon dioxide, glycerine was used for some time, but now a mineral oil is used exclusively in the practice in the United States and Great Britain. On account of the low temperatures so often encountered with carbonic compressors, a much lower test oil is required than for ammonia. Chemically pure, dry glycerine has been considered to be the only practical lubricant for ethyl and methyl chloride, whereas sulphur dioxide requires a straight refined, highly anhydrous, mineral oil. The hydro-carbon liquids tend to dissolve mineral oils, and where the liquid gets into contact with the oil to any extent it is found that glycerine or one of the glycols mixed with deflocculated graphite will give good results.

Tables and Charts.—Figures 164 to 168, inclusive, are P–I diagrams for carbon dioxide, ammonia, ethyl chloride and methyl chloride respectively. These diagrams (see Chapter I) represent the action of the entire refrigerating cycle and they present an easy and accurate method for the solution of problems in refrigeration. Tables 59, 61, 62 and 64 were compiled by the Linde Air Products Company.[9] Table 56 and Fig. 166 are from D. L. Fiske, Refrigerating Engineering, Dec., 1924. The properties of saturated and superheated ammonia, and the P–I diagram for ammonia are taken from Circular 142 of the Bureau of Standards.

[8] W. F. Osborne, Journal of the Amer. Soc. Refrig. Eng., 1920.

[9] Refrigerating Engineering, June, 1926.

CHAPTER VIII

BRINE AND BRINE SYSTEMS

As before noted, refrigeration may take place *directly* through the absorption of heat by the refrigerant from the commodity to be cooled, and *indirectly* where water or some non-freezing solution is cooled by the refrigerant and then in turn cools the commodity by means of pipe coils or by sprays. The small job generally uses the direct method (called *direct* expansion), while for larger work whether the direct or the indirect (the brine system) shall be used depends on a number of factors, so that each case has to be decided separately on its merits.

Costs.—With direct expansion piping for ammonia it is necessary to have an extremely tight system. With steam and compressed air piping a small leak is usually of little moment, but in the case of ammonia even a small leak may be very serious, both as regards the cost of the refrigerant lost and also because of the possible danger to persons and commodities from the released ammonia. To keep the pipe line tight means special fittings and very careful pipe work, with either the tinned and soldered or the lithage and glycerine thread joint.

On the other hand, the brine refrigerating piping need not be special. The pressures used are generally not greater than 25 lb. in excess of the static pressure due to the head of brine, and the piping and the fittings can be the usual kind for 125 lb. pressure. If the brine system is a closed one with a balance tank (Fig. 169), the pumping cost need be only that which is required to overcome friction and create the velocity head. There is required, however, some kind of brine cooler of the double pipe, the shell and coil, but usually of the shell and tube variety, and the necessary circulating pump for the brine. As the brine is used to *absorb* heat and its heat capacity is limited to a rise in temperature of a few degrees only, it follows that the brine circulating pump must be in operation as long as refrigeration is required, unless hold-over or congealing tanks or brine tubes are used to hold a quantity of brine and thereby store up a relatively small amount of refrigeration.

Direct expansion means, in plants of any size, a heavy initial cost for the charge of ammonia or other refrigerant. Should a large leak occur, due to a broken fitting or pipe, there is danger of the entire charge being

lost. With brine it is also possible to lose the charge, but if the piping is open to inspection there should not be much difficulty in keeping the loss small.[1]

The refrigerating machine can be operated periodically when brine and a brine storage tank are used, but the brine pump needs to operate continuously. In direct expansion the compressor has to pump as long as refrigeration is desired.

Advantages of Brine.—There is an advantage in the use of brine from the fact that it may be used as a storage for refrigeration and therefore the brine-operated plant with brine storage can respond to sudden demand, as, for example, in the quenching of steel, in the pasteurizing of milk or in other cases where the cooling required comes in

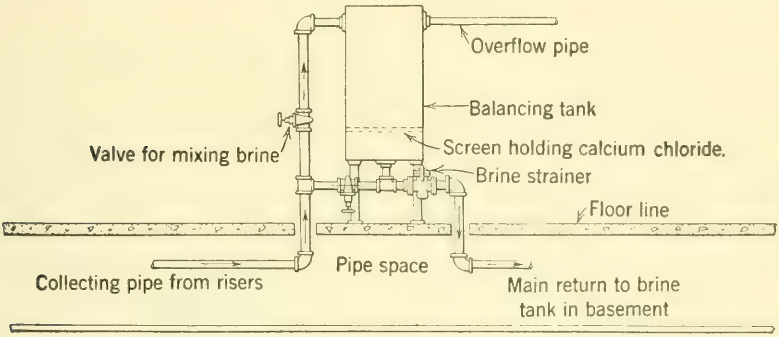


FIG. 169.—A Typical Connection to a Brine Balancing Tank.

sudden demands. Under these conditions it is possible to use a brine storage tank of size sufficient to obtain the required storage capacity, in which case a smaller refrigerating machine may be operated practically continuously. When the temperatures required are not below the temperature of freezing of water (32 deg. F.), it is usual to substitute water for the brine, unless (as in the case of steel quenching) there are good reasons for the use of calcium or sodium chloride brine. Some kinds of work are best performed by employing a spray (particularly in the case of air cooling and conditioning), using a specially designed spray nozzle so as to atomize the brine or water as finely as possible. The special advantage here is in the washing of the air, and—for the lower temperatures—in the condensation of the moisture in the air so as to make the moisture content correspond to the dew point temperature without the freezing of the moisture on the pipes. It has been found also that a quicker and more satisfactory cooling can be obtained

[1] See District Cooling, Chapter XIV.

at times by the use of sprays[2] due to the intimate contact of the spray with the air. For example, in the packing house where the hot carcasses must be cooled quickly, uniformly, and with least shrinkage in the weight, it has been found that the brine spray is the best to be used. The spray deck is designed so that the entire air volume of the room in which the hot meats are hung can be changed (or recirculated) once a minute or once per one-half minute. The sprays drag the air in the direction of their flow and the nozzles are so placed as to provide a curtain of liquid through which the air has to pass. The air is thereby cooled to the temperature of the brine or of the water. (See Chapter XIV.) Water sprays are made use of extensively in the cooling of theatres and auditoriums, in chocolate and gelatine factories, or whereever a temperature of 35 degrees or more can be employed with satisfaction.

There are some industries which cannot permit the use of ammonia in the process plant. Carbon dioxide usually is innocuous, but carbon dioxide is not in frequent use in the United States. In England and in Northern Europe the condensing water is usually quite cool and the result is that carbon dioxide can be used with satisfaction, and is used for about 50 per cent of the time, whereas in India and other tropical countries with water temperatures in the neighborhood of 100 deg. F. or higher the refrigerant may be sulphur dioxide or ethyl chloride. Where leaks of ammonia may cause considerable damage, and careless labor makes the danger of accidents a live one (as is the case in some industrial plants) it may be best to use the indirect system, employing of brine or refrigerated water. However, it must be kept in mind that the indirect brine system is always operated at greater expense, as in cooling brine for use in refrigerating piping the boiling temperature of the ammonia must be, usually, 10 deg. F. lower than the boiling temperature of the ammonia when the direct expansion piping is used. This is because of the double cooling that is required, although when sprays are used it is doubtful whether this statement is strictly correct. The brine has to be circulated (which also assists in heating the brine by an amount equal to the heat equivalent of the power used), and at times this power item is an important one. In direct expansion the liquid ammonia will pass to the expansion valve because of its own static pressure due to the condenser pressure, and the suction line, which usually is relatively short, is designed so as to reduce the pipe friction loss to a minimum. Usually it is necessary to restrict the length of

[2] With brine sprays, especially if the temperature is low, foaming may result, in which case the surface tension of the brine needs to be increased. This can be done by the addition of a little kerosene plus oleaic acid.

the suction return line, and so when the compressor has to be remote from the location of the process requiring cooling, the brine system has to be used.

Brine Cooling.—In cooling brine there is a decided advantage in using the shell and tube brine cooler (Figs. 170 and 171). When well

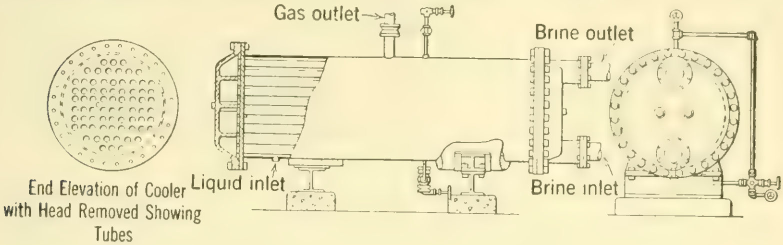


FIG. 170.—Shell and Tube Brine Cooler for Eight Passes.

designed and constructed it is simple, compact and durable, and large capacity is obtained with efficiency of surface and with little trouble from leaks. As a rule a surface of 14 to 15 sq. ft. per ton of refrigeration is used. The construction is very similar to that of the steam surface

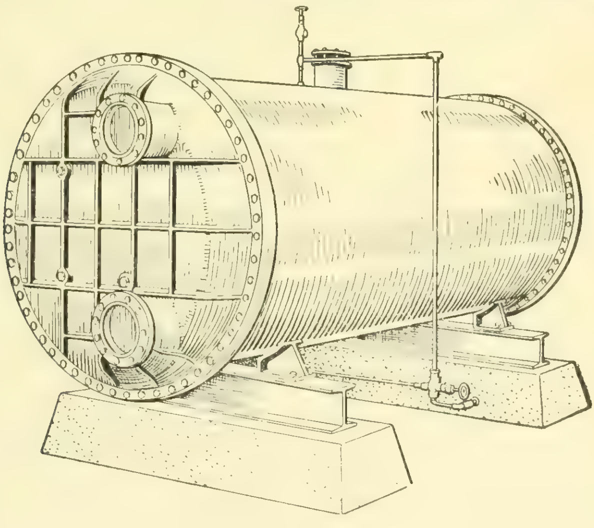


FIG. 171.—Shell and Tube Brine Cooler.

condenser, and the horizontal return tubular boiler. The diameter is usually limited to from 48 to 54 in., and the shell thickness varies from $\frac{1}{4}$ to $\frac{7}{16}$ of an inch, depending on the diameter of the shell, while the tube sheet is usually $\frac{1}{2}$ in. The tubes are usually 2-in. normal diameter.

and are of No. 12 gage charcoal iron, and are expanded into the tube sheet. The design may permit a single pass of the brine, as is the case of the ice-making installations, where the cooler is submerged in the brine tank, or there may be eight or more passes by the use of baffles in the heads. As a rule the number of passes should be kept down to six, for one of the advantages of the use of the brine cooler is in the high value of the coefficient of heat transfer which is due in part to the rapid flow of the brine through the tubes. If the number of passes becomes too great, the resistance to the brine flow may become excessive and costly. Examples are known where the pressure difference between the entrance and the exit has been as much as 75 lb. per sq. in.

Shell and tube coolers are liked because the ammonia part may be made very self-contained. With the possible exception of the condensers (which may be on the roof), the whole system may be placed in the

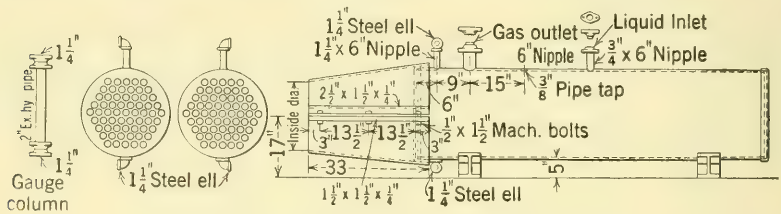


FIG. 172.—Single Pass Brine Cooler.

compressor room under the eye of the engineer. The ammonia charge is not so great and the pressure drop from the evaporator surfaces can be made almost inappreciable. Only one expansion valve need be used (per cooler), and the liquid is best carried at about three quarters full. A liquid trap can be placed in the suction line about six feet above the top of the cooler to separate the liquid ammonia entrained in the suction gas, and this liquid may be returned to the cooler through a pipe line, using a goose-neck connection.

The advantages of the horizonal over the vertical shell and tube cooler are as follows: The ammonia in the horizontal cooler does not have the free passage out as does the vertical one, and the liquid priming is caught by the tubes which are staggered to some extent. The head-room required is not so great, although more space is required on the floor, but there is greater ease in any repair work that is necessary. These coolers should have connections at the top and the bottom for the gage glass, an outlet at the bottom for a blow-off and the liquid ammonia inlet, and at the top for the gas outlet connection. The shell and the tube brine cooler has the disadvantage of danger due to the freezing of the

brine, and the consequent possible bursting of the tubes. It is advised that the calcium chloride brine only be used, because of the much lower freezing temperature of this brine, and that a careful check be kept of the density. Even under these conditions the stoppage of the brine pump may cause trouble, if the ammonia boiling temperature is such as will be below the freezing temperature of the brine.

The design of the brine pipe system (Figs. 174, 175, 176, and 177) is not very well standardized. The essential feature is to reduce the pumping cost to a minimum. This means, wherever possible, the use of an expansion tank at the high point and of a piping arrangement so

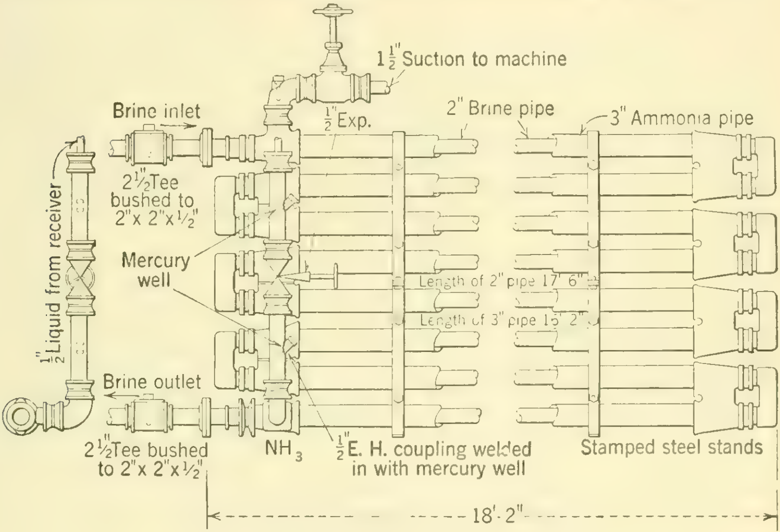


FIG. 173.—Double Pipe Brine Cooler.

that the brine operation in a complete circuit will be of the nature of a syphon. Under these conditions the pump work consists only of the friction and velocity head. The static pressure on the pump, of course, is due to that required to overcome friction plus the head of brine from the pump to the expansion tank.

Corrosion of Brine.—Considerable attention has been directed towards the causes, and the means of their reduction, of corrosion in the brine system. Corrosion may be the result of a number of causes due to the fact that all brines are electrolites, which under certain conditions may stimulate galvanic action. This galvanic action may be caused by dissimilar metals or even by the difference in the potential in different parts of the same metal due to stresses developed in the manufacture or the assembling of the material. Mill scale will sometimes

start corrosion as will also "stray" direct currents on their return to the generator in the power plant, or the presence of air in solution in the liquid may be the cause of the trouble.

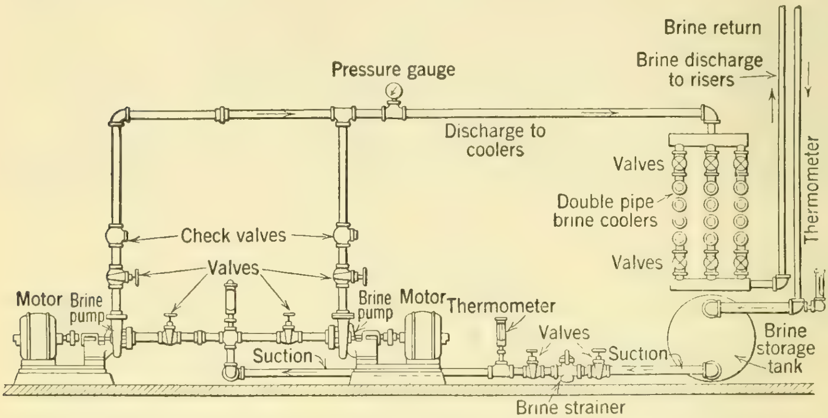


Fig. 174.—Brine Cooler, Brine Tank and Pump.

From the preceding it is evident that if corrosion is to be reduced to a minimum these factors must be removed or decreased in relative importance, and such brine should be used as will give the best results for the work being carried on. At present there are three brines in general

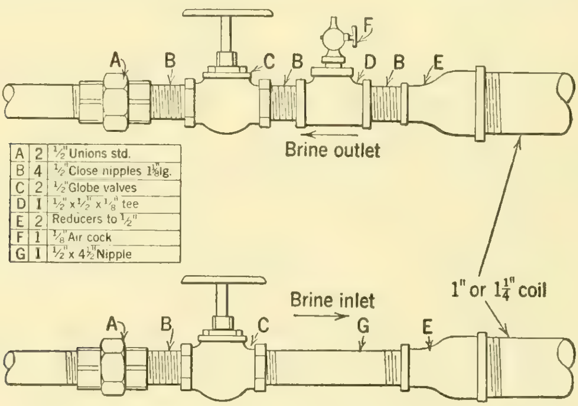


Fig. 175.—Typical Connections to Brine Coils.

use: the sodium or salt brine, the calcium chloride and the mixed magnesium-calcium chloride brines. Which of these different kinds of brines shall be used depends on the kind of service to be rendered. For

example, the packers using spray nozzles to a large extent in the chill rooms for chilling the warm carcasses employ salt brine practically entirely for that service, because of the greater "drying" action of calcium, its greater cost, etc. Calcium chloride is used particularly where low temperatures are desired, or where it is thought that there is danger of the freezing of the brine cooler, in which case there is greater safety in using a brine of lower freezing temperature than is possible with salt brine.

According to tests by Morgan B. Smith,[3] all brines should be tested

[3] Morgan B. Smith, A. S. R. E., 1911:

"The protection of refrigerating apparatus against corrosion involves three main heads, the metallic apparatus, the brine solution and the relation of the refrigerating apparatus proper to the other apparatus in the vicinity, especially with regard to the generation of electrical current. With regard to the refrigerating apparatus, several factors must be considered, i.e., first, the nature of the metals used with consideration of their ability alone to resist corrosion, and their tendency in combination to set up galvanic action and induce corrosion; second, the design of the apparatus and the general workmanship in assembling the varied elements which go to form the finished plant.

"The nature of the metals used and their combination is worthy of very close study as both practice and experimentation have proved. For example, if we place two electrically dissimilar metals, such as brass and iron, in a brine solution an electric current is at once set up—such a current passing from the iron to the brass through the brine solution. The action of this current is such that the iron gradually passes into solution and subsequently is oxidized to form the familiar oxide called rust. The solution of the iron in such a case may take years. On the other hand, it may take a much shorter time, for the rapidity with which such a reaction takes place varies directly with the magnitude of the current set up. This is equally true of the combination of any electrically dissimilar metals, only varying with the relative degree of dissimilarity in the metals used, and for this reason it is perfectly obvious that but one metal should be used as far as possible in refrigerating plants.

" Not only should but one metal be used as far as possible but that metal should be of the greatest possible purity, for impurities in metals act in much the same manner as do two dissimilar metals mechanically combined. This is especially true of cast iron, which is frequently destroyed by the numerous local currents set up between the iron and its impurities leaving the casting in a characteristic graphitic condition. It is well at this time to consider a widely used combination of two electrically dissimilar metals, namely, galvanized iron.

" Zinc resists corrosion in a brine solution if no electrically dissimilar metal be present. If, however, iron is present the zinc will go into solution, thereby protecting the iron as long as zinc is present in sufficient amounts. Therefore it is plain that galvanized iron should never be used excepting when of the very best quality, in which no iron is present that is exposed to the brine. When galvanized iron does show iron exposed at pin holes, or along seams where bending has cracked the zinc film it is next to worthless in brine solutions, for sooner or later all the zinc will be stripped off leaving the iron bare. As there is evidence that the highest grade of iron is not used in the manufacture of galvanized iron, it is plain that the iron itself will be more liable to corrosion than would a plain iron apparatus of a high grade of purity. It is doubtful, therefore, if galvanized iron ever should be used since it seems almost impossible to construct apparatus of this material so that only zinc will be exposed to the brine solution.

" Many alloys resist corrosion to a marked degree, especially the bronzes. Alloyed copper in iron tends to retard corrosion, whereas unalloyed copper in iron greatly hastens corrosion. (Eng. News, Aug. 3, 1911.) From the foregoing facts it is very evident that as far as practical one homogeneous metal only should be used in the construction of apparatus.

" The design of apparatus, especially of fittings and piping, is very important and is generally wholly neglected. When a piece of metal is strained an electrical difference of potential is at once set up between the strained and the unstrained portions, and it is the strained portion which is attacked and eventually destroyed. This state of affairs is exemplified in the case of threaded pipe ends, engaging fittings, the pipe ends being strained while the remainder of the pipe is unstrained. There can be no doubt whatever that this state of strain in pipe ends is largely responsible for the failure of piping at the threaded portions. This is almost equally true of fittings which,

from time to time for acidity, as acidity increases very greatly the process of corrosion. In general he advises the use of some alkali like lime (CaO) or caustic soda, or the use of a bag of lime suspended in the tank or at some other convenient location. In his opinion, "the least corrosive chloride is that *calcium chloride* which contains as small an amount of magnesium and sodium chloride and the least practical amount of water." He also says that there is no longer any doubt that magnesium and sodium chloride increase corrosion when present in

however, generally have so much more metal remaining after the threads are cut that the strain caused in them when making up joints cannot be so great as it is in piping where comparatively little metal is left after threading. There are a number of instances where pipe trouble has been eliminated by the use of piping and fittings of extra weight, so that after the threads are cut there is left an abundance of metal sufficient to withstand the making up of joints without undue strain.

"Air is one of the great accelerators of corrosion, and it is very essential to see that no air pockets can form in the apparatus, especially in the piping. All high points in the piping and each piece of apparatus as far as practicable should be provided with suitable vents so that any trapped air can be withdrawn from time to time. If an air pocket does exist for some time, corrosion soon makes itself evident, and if allowed to go on will eventually cause failure at the point.

"We may now consider the brine solution to be used. Since the brine is made up by dissolving certain chlorides in water, it is very evident that the nature of the water is sure to affect the resulting brine solution. Water containing acid of any description should never be used, since the resulting brine will also be acid. An acid brine is far more corrosive than an alkaline or a neutral brine.

"As a matter of fact all brine solutions should be tested from time to time for acidity, and if found to be acid should be neutralized at once with some alkali such as lime or caustic soda. Sufficient alkali should be added to make the brine distinctly alkaline towards a suitable indicator. The best indicator for such purpose is *phenol-phthalein.* This is made up by dissolving phenol-phthalein in alcohol—about one-half ounce in two quarts—and adding to the solution two quarts of distilled water. Another good indicator which may be used is methyl-orange, which turns yellow on adding brine if the brine is alkaline and red if the brine is acid. When making up the brine it is well to avoid saturating the brine with air, for brine saturated with air is far more corrosive than brine carrying only a minimum amount of air in solution. Brine should never be agitated with an air-blast, nor should it be allowed to fall through the air into tanks as is often the case with return brine. Return brine is often very nearly saturated with air in just this manner. As for the strength of the brine and the temperature it is certain that the greater the concentration and the lower the temperature the less likely is the brine to assist corrosion.

"The selection of chloride to use in making up the brine solution is no longer a serious problem since practice and experimentation have proved the truth of the assertion that the most economical and the least corrosive chloride is that calcium chloride which contains as small an amount of magnesium chloride and of sodium chloride as possible, and the least practicable amount of water.

"There is no longer any doubt that magnesium chloride and sodium chloride in calcium chloride brine materially increase the corrosiveness of the resulting brine, compared with a brine containing practically no chloride other than calcium chloride. This is especially true if the brine containing magnesium chloride becomes acid through the unstable nature of magnesium chloride.

"I have been asked whether ammonia present in brine would increase the corrosiveness of the brine. Experience of long duration, in which the ammonia content of the calcium chloride brine in question was maintained at a constant value, shows that *ammonia tends to retard rather than to hasten it,* when present in sufficient amount.

"In addition to the effect of galvanic action caused by different materials in the system corrosion may be greatly hastened by the action of stray electric currents which find a way into the refrigerating apparatus from which they pass to some neighboring metallic circuit or directly to the ground. If the refrigerating apparatus, in whole or in part, is positive in electrical potential towards the ground we may be sure that some stray electric current is finding its way into the apparatus, and that corrosion is being hastened in proportion to the magnitude of the stray current. It is said that a difference of potential of 1/1000 volts is all that is required to start corrosion two miles from the dynamo. It is a wise precaution, therefore, to see that direct current apparatus of all sorts is properly insulated so that it will not be a continual menace to the refrigerating engineer and his plant."

calcium brine. This, he says, is on account of the unstable nature of magnesium chloride, which tends to become acid and cannot be made alkaline.

Emerson P. Poste[4] says, as a result of his own experiments, "essentially pure calcium chloride is alkaline when first made but it turns acid on exposure to air. With this acidity comes a marked increase in the corrosive action. The acidity is materially increased by the presence of magnesium chloride, as the result of the earlier development of acidity and the formation of corrosive ammonium chloride in the case of ammonia leaks. Contact of unlike metals and the presence of these impurities accelerate these electric tendencies. The corrosive action of brine decreases with increasing brine density. Chlorides are on the market which are contaminated with magnesium chloride, though first-class materials are available both as soda by-product and natural chlorides. Corrosive brines free from magnesium chlorides may be corrected by treatment with lime if the alkalinity produced is maintained above 0.1 per cent.

"The careful plant manager who

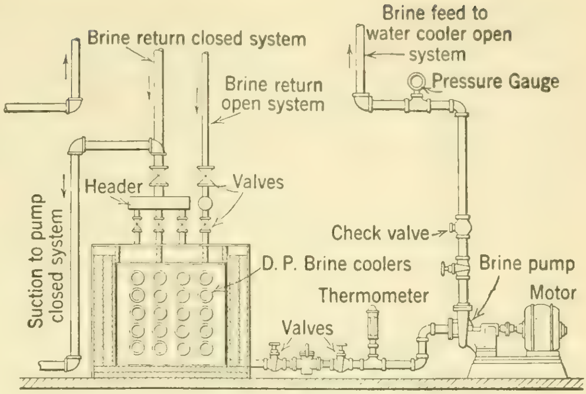


FIG. 176.—Typical Connections to Brine Pump.

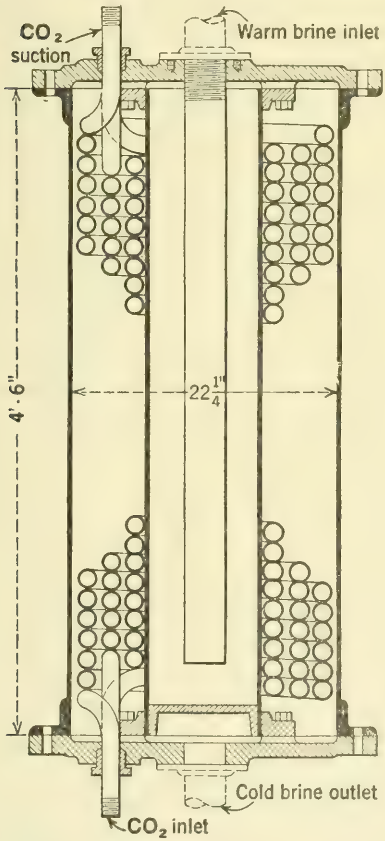


FIG. 177.—Shell and Coil Brine Cooler for Carbon Dioxide.

[4] Emerson P. Poste, Milk Dealer, January, 1923.

is interested in avoiding operating irregularities, in keeping his equipment in the best of condition and in getting the most out of it will use only the best of chloride in making his brine, will keep the gravity approaching 1.2, will hang a bag of lime in his brine tank and maintain therein a good supply of material, and will avoid unnecessary contact between the brine and the air." Fig. 178, taken from these experiments, shows the relative rates of corrosion.

The presence of ammonia in the brine caused by ammonia leaks frequently causes considerable concern other than because of the loss of the ammonia. Morgan B. Smith claims that experiments of long duration, during which time the ammonia content was maintained constant, show that ammonia tends (as has been said before) to retard rather than to hasten corrosion. Of course, in the case of ammonia pipe leaks, the thing to do is to stop the leaks. Much ammonia in the brine will probably remove the galvanizing from the cans (if the brine is in an ice tank), but after this is removed the corrosion will stop unless ammonium chloride is formed as it will if magnesium chloride is present. Sal ammoniac will accelerate corrosion of iron and steel. If ammonium chloride is not formed it would not seem that any great harm is caused by the presence of an alkaline brine occasioned by the presence of ammonia.

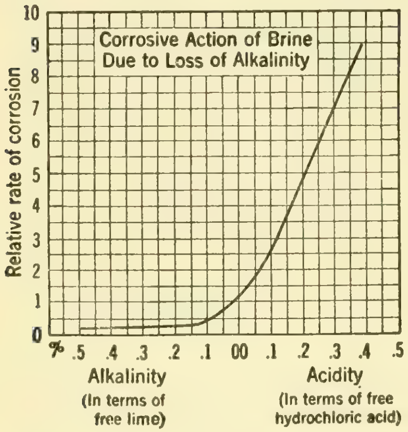


FIG. 178.—Relative Corrosion of Brine

Too much alkalinity is undesirable where galvanized coatings form a large part of the metal exposed to its action,[5] as would be the case in can ice freezing tanks. Experiments prove that commercial calcium chlorides now on the market show decided corrosive action due to their high soluble alkalinity, but weak brines show more action than do the stronger brines of the same degree of alkalinity. Weak brines with ammonia or ammonium chloride present tend to increase the amounts of corrosion.[6]

[5] Arthur C. White, Industrial and Engineering Chemistry, May, 1925.

[6] As the ice can is always galvanized, galvanic action is always present if the brine is acid. In this case it has been shown by A. C. White that "the action of calcium chloride brine on galvanized coatings is such that the galvanizing can be saved by having the least alkalinity possible, and that neutral brines have the least effect."

Should, however, it be decided to remove the ammonia from the brine it is most probable that heating the brine will be found to be the only practical method. In the larger plants, where care is taken in details, and the brine is kept alkaline and not permitted to get weak or to absorb air, corrosion should not be great with any kind of brine, and at times the cheap brine (if much brine has to be wasted) with some corrosion will be found to be more economical than the more expensive kind that is not so corrosive.

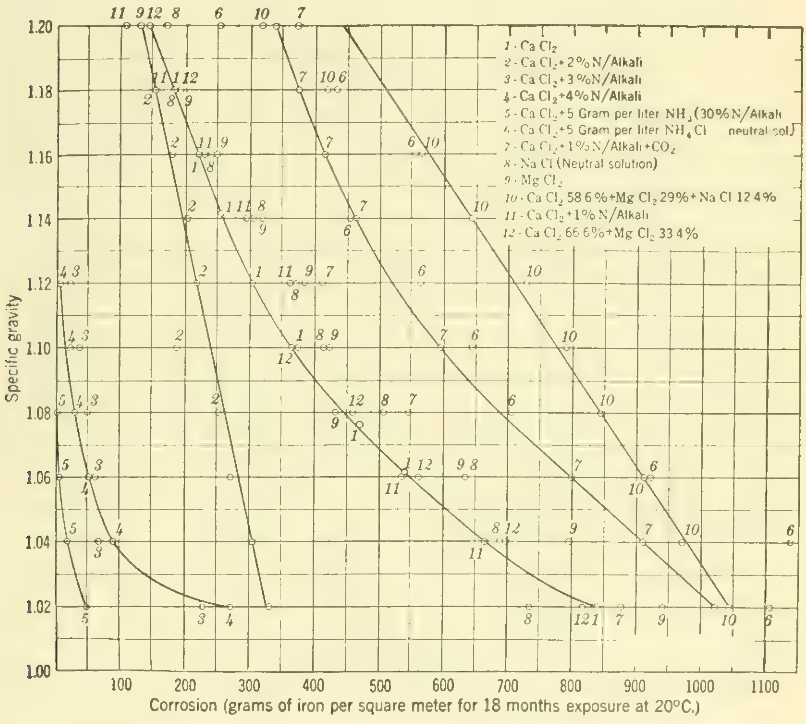


FIG. 179.—Tests on Corrosion of Calcium Chloride Brine.

Figure 179 shows the results of tests by C. A. Cole in the laboratory of the Senet-Solway Company. Twelve series of experiments were run at the same time, each series being made up of 10 tests under the same conditions. In each test a 3-in. piece of $\frac{1}{4}$-in. polished iron taken from the same rod was completely submerged in 100 c.c. of solution. The iron had an analysis of 1.17 per cent C, Mn and Si nearly $\frac{1}{10}$ per cent, and S 0.06 per cent. These results showed that corrosion decreased with the concentration. The solution should be alkaline and should not be exposed to air or to CO_2 in the air. As magnesium chloride solution

cannot be made alkaline, such solutions are objectionable because of the increased corrosion. Finally, brine solutions should not contain any considerable amounts of three constituent salts.

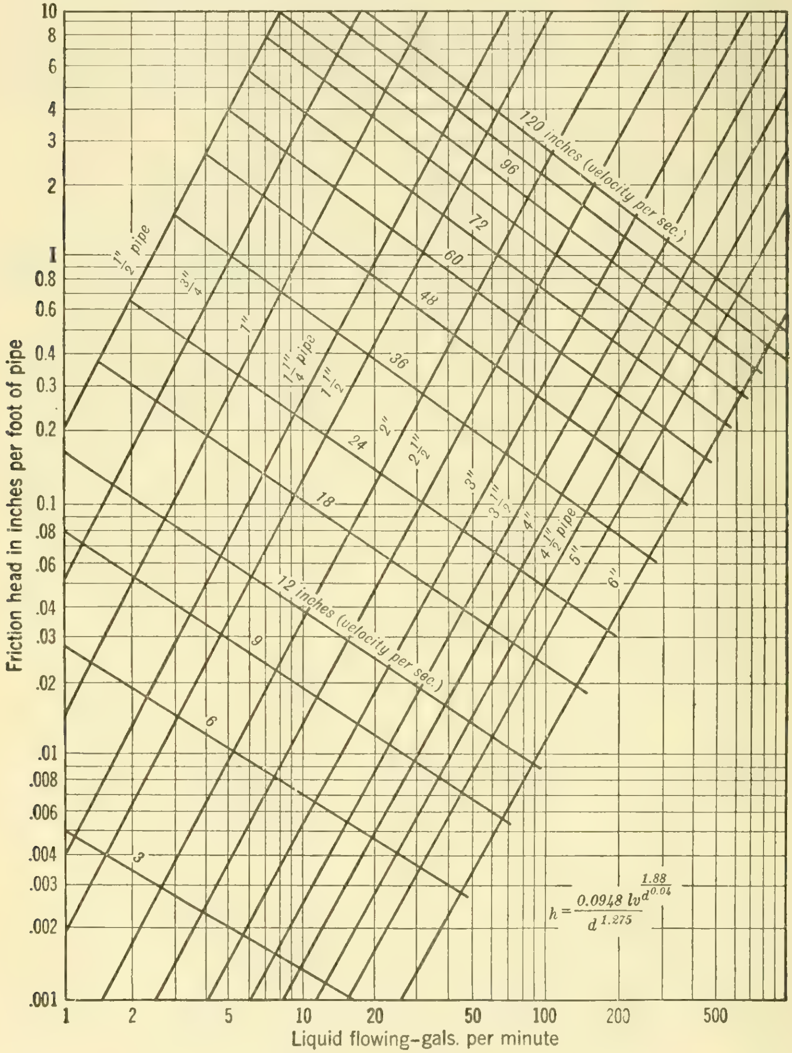


FIG. 180.—Resistance to Flow of Water in Pipes.

Report of the Corrosion Committee.—The report of the Corrosion Committee[6a] is abstracted as follows:

[6a] Report of the Corrosion Committee, Refrigerating Engineering, Jan., 1927, by Russell, Roberts and Chappell.

For Closed Brine Systems.– Use may be made of sodium dichromate ($Na_2Cr_2O_7 + 2H_2O$) and sufficient sodium hydrate to convert the dichromate to chromate. For calcium chloride brine use 100 lb. of dichromate per 1000 cu. ft. of brine and add 35 lb. of 76 per cent caustic. For sodium brine use 200 lb. of dichromate per 1000 cu. ft. of brine and add 70 lb. of 76 per cent caustic. The brine should be slightly alkaline after adding the salt and the caustic.

Open Brine Systems.– For sodium chloride brines use di-sodium phosphate ($Na_2HPO_4 + 12H_2O$) at the rate of 100 lb. per 1000 cu. ft. of brine. The brine must not be too alkaline. Test for alkalinity with phenolphthalein and if it shows *pink* hydrochloric acid is to be added. Test strips of bright iron or steel should be suspended in the brine in. order to note the rate of corrosion.

Zinc Dust.—Zinc dust may be used to advantage in open calcium chloride systems at the rate of 60 lb. of zinc dust per 1000 cu. ft. of the brine. A little should be added at a time.

Condensers.—For fresh water recirculating condenser systems use 17 lb. of 40 degree silicate per day per 1000 cu. ft. of make up water. The water should test pink with phenolphthalein one hour after adding the silicate.

Friction to Flow of Brine.—The laws underlying the resistance to flow of brines have been given very little attention. The only published information on the subject is a paper by Professor A. H. Gibson[7] of the

[7] A. H. Gibson, The Resistance to the Flow of Brine Solutions through Pipes, The Institute of Mechanical Engineering, Feb., 1914.

Lord Raleigh has shown that the resistance to flow of a non-compressible fluid can be given by:

$$R \propto \rho v^2 \phi\left(\frac{\mu}{\rho v d}\right) \text{lb. per sq. ft. of surface,}$$

where

ρ = the density of the fluid in pounds per cubic foot;
μ = coefficient of viscosity of the fluid in foot-pound units;
v = mean velocity of flow;
d = diameter of the pipe;

$\phi\left(\frac{\mu}{\rho v d}\right)$ must be found by experiment.

Osborne Reynolds found by experiment for water flowing in pipes

$$R \propto \rho v^2 \left(\frac{\mu}{\rho v d}\right)^{2-n} \text{lb. per sq. ft. of surface.}$$

Footnote continued on page 266.

University College, Dundee. Dr. Gibson states that, as pointed out by Reynolds, with a rough pipe the frictional head loss is independent of

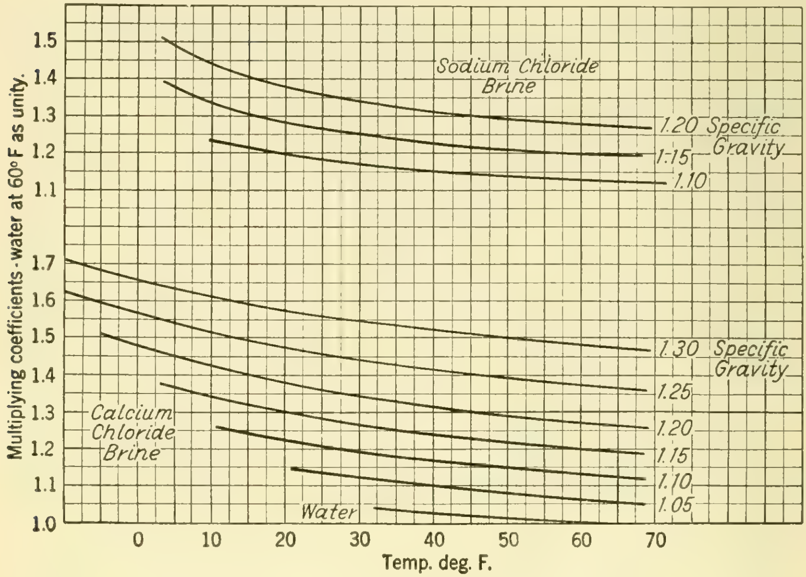


FIG. 181.—Gibson's Multiplying Factors for Brine.

viscosity and that for velocities above the critical the resistance to flow is proportional directly to the densities. Using a Reynolds' number of

Putting h = the loss of head in a pipe of length l, expressed in terms of feet of a column of the fluid flowing then as

$$F \propto R \times \text{the wetted surface,}$$

$$h\rho\frac{\pi d^2}{4} \propto \rho v^2\left(\frac{\mu}{\rho v d}\right)^{2-n}\pi dl,$$

or,

$$h \propto \left(\frac{\mu}{\rho}\right)^{2-n}\left(\frac{v^n}{d^{3-n}}\right)l \text{ ft.}$$

Therefore different fluids with identical values for the index n will be related according to

$$\frac{h_1}{h_2} = \left(\frac{w_1}{w_2}\right)^{2-n}\left(\frac{v_1}{v_2}\right)^n, \quad \text{where} \quad w = \frac{\mu}{\rho},$$

and at identical velocities,

$$\frac{h_1}{h_2} = \left(\frac{w_1}{w_2}\right)^{2-n}.$$

1.9 for all pipes, Dr. Gibson calculated multiplying factors for salt and calcium brine. These factors are to be the multipliers of the friction head of water at 59 deg. F. The head lost by water is shown in Fig. 180, and the multiplying factor is given in Fig. 181. However, the results given in these curves appear to be large by an amount varying from 10 to 20 per cent.

Concrete Brine Tanks.—The concrete brine tank can be used to advantage at times. According to Boyden,[8] a mixture of 1 : 2 : 4 under 30 ft. of water is waterproof. The concrete must be made of fine clear sand and hard, durable crushed rock or screen gravel, graded from $\frac{1}{4}$ in. to 1 in. The water should be pure and about 6 gal. per sack of Portland. Use a mixture of 1 : $1\frac{1}{2}$: 3, and if possible pour all at once. The inside should be finished in rich mortar rubbed in with a brick. The tank must be kept free of other construction and the ground underneath should be guarded against freezing.

[8] H. C. Boyden, Journal A. S. R. E., January, 1921.

TABLE 65

Capacities of Double Pipe Brine Coolers, $1\frac{1}{4}$-In. and 2-In. Pipe, Tons of Refrigeration

Pipes High	Brine Range																							
	2						3						5						10					
	Mean temperature difference																							
	5	7½	10	12½	15	17½	5	7½	10	12½	15	17½	7½	10	12½	15	17½	20	10	12½	15	17½	20	22½
	Capacity of cooler 10 ft. long over all																							
2	....	.11	.33	.55	1.1	1.6	....	....	.08	.20	.48	.64	....	....	....	.13	.16	.27	....	....	....	....	.11	.13
3	.11	.45	1.0	1.4	2.2	2.8	....	1.6	.48	.90	1.3	1.8	....	....	.27	.40	.67	1.3	....	....	....	.20	.27	.40
4	.28	1.0	1.6	2.3	3.1	4.4	.12	.40	1.1	1.6	2.4	2.9	....	.40	.67	1.3	1.6	2.5	....	....	.27	.53	.80	1.1
6	.83	2.0	3.0	....	....	....	.48	1.2	2.2	3.2	4.4	....	.54	1.3	2.3	3.2	3.7	5.1	.27	.53	1.1	1.6	2.7	3.7
8	1.5	3.0	....	....	....	....	1.1	2.2	3.5	....	....	....	1.3	2.4	3.6	4.9	....	....	.53	1.6	2.7	4.0	4.8	6.4
10	2.2	....	....	....	....	....	1.6	3.2	....	....	....	....	2.3	3.6	5.3	....	....	....	1.3	2.8	4.5	5.8	7.2	8.8
12	....	....	....	....	....	....	2.2	....	....	....	....	....	3.1	4.9	....	....	....	....	2.6	4.5	6.1	8.0	9.9	
14	....	....	....	....	....	....	....	....	....	....	....	....	4.0	....	....	....	....	....	4.0	5.8	8.0			
16	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	5.1	7.5	...			
18	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	6.1	8.8				
20	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	7.5					
	Gallons per ton per minute, 1.25 specific gravity of brine																							
	18						12.5						7.5						3.75					
	Capacity of cooler 20 ft. long over all																							
2	.39	1.0	1.4	2.2	3.6	5.0	.12	.56	1.3	2.0	2.6	3.2	.16	.40	.90	1.6	2.3	2.9	.13	.26	.33	.53	.80	1.3
3	1.1	2.2	2.5	....	....	....	.56	1.6	2.6	3.7	4.9	....	.67	1.6	2.6	3.6	4.6	5.6	.26	.66	1.3	2.1	3.2	4.2
4	1.8	3.4	....	....	....	....	1.2	2.6	4.1	....	....	....	1.6	2.9	4.2	5.6	7.3	....	.80	1.8	3.2	4.5	5.8	7.2
5	2.5	....	....	....	....	....	1.9	4.1	....	....	....	....	2.7	4.2	6.1	....	....	....	1.8	3.1	5.2	6.9	8.5	10
6	....	....	....	....	....	....	2.6	....	....	....	....	....	3.6	5.7	....	....	....	....	3.2	5.1	7.2	9.3	12	
8	....	....	....	....	....	....	....	....	....	....	....	....	5.7	....	....	....	....	....	5.8	8.5	12			
10	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	8.5	12				
12	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12					
14																								
16																								
18																								

TABLE 66

Capacities of Double Pipe Brine Coolers, 2-In. and 3-In. Pipe, Tons of Refrigeration

Pipes High	Brine Range																							
	2						3						5						10					
	Mean temperature difference																							
	5	7½	10	12½	15	17½	5	7½	10	12½	15	17½	7½	10	12½	15	17½	20	10	12½	15	17½	20	22½
	Capacity of cooler 10 ft. long over all																							
2	.04	.08	.16	.39	.78	1.4	.04	.06	.10	.16	.24	.40	.04	.03	.11	.13	.20	.27	...	...	.08	.13	.19	.19
4	.14	.55	1.2	2.0	3.4	6.6	.10	.20	.56	1.1	1.8	2.6	.13	.20	.38	.67	1.1	1.6	.05	.16	.26	.33	.40	.53
6	.67	1.7	3.3	4.9	8.0	...	.24	.96	2.0	3.3	4.8	6.4	.33	.80	1.6	2.3	3.6	5.0	.26	.40	.67	.94	1.6	2.1
8	1.3	3.1	5.4	7.9	...	...	.64	1.9	3.7	5.4	7.7	8.4	.80	1.7	3.2	4.9	6.7	8.7	.40	.80	1.6	2.4	3.5	5.1
10	2.2	4.4	7.9	...	...	...	1.2	3.2	5.6	8.2	...	...	1.5	3.2	5.2	7.7	10	13	.80	1.6	2.9	4.5	6.4	8.5
12	3.2	6.1	...	...	...	...	1.9	4.6	7.6	...	...	...	2.4	4.8	7.7	10	13	...	1.3	2.9	4.8	6.9	9.6	13
14	4.7	...	...	...	...	...	3.0	6.7	...	...	...	...	4.0	7.5	11	14	...	...	2.9	5.1	8.0	12	15	19
16	5.3	...	...	...	...	...	3.7	7.7	...	...	...	...	4.8	10	13	...	...	...	3.5	6.2	9.6	13	17	21
18	...	...	...	...	...	...	4.6	...	...	...	...	...	6.3	...	...	...	...	...	4.5	8.3	13	17	21	
20	...	...	...	...	...	...	...	...	...	...	...	...	7.7	...	...	...	...	...	6.1	11	16	20		
	Gallons per ton per minute, 1.25 specific gravity of brine																							
	18						12.5						7.5						3.75					
	Capacity of cooler 20 ft. long over all																							
2	.44	. .	1.8	2.8	3.6	5.0	.12	.64	.88	1.6	2.5	3.6	.16	.27	.53	. .	1.7	2.4	.19	.27	.32	.43	.53	.80
4	2.5	3.9	6.3	...	...	...	.88	3.6	4.5	6.7	9.0	1	1.0	2.4	4.2	6.0	8.2	10	.53	1.1	2.0	3.5	4.8	6.4
6	3.7	7.5	...	...	...	...	2.4	5.4	8.8	...	...	...	3.1	6.0	8.9	13	16	...	1.9	3.7	6.1	8.8	12	15
8	...	...	...	...	...	...	4.5	8.8	...	...	...	...	6.0	10	15	...	...	...	4.5	8.0	12	16	20	25
10	...	...	...	...	...	...	6.6	...	...	...	...	...	9.1	15	...	...	...	...	7.7	13	18	23	27	
12	...	...	...	...	...	...	...	...	...	...	...	...	13	...	...	...	...	...	12	18	25			
14	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	15	22				
16	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	18					
18																								
20																								

TABLE 67

CAPACITIES OF MULTIPASS SHELL AND TUBE BRINE COOLERS—FLOODED, TONS OF REFRIGERATION PER 24 HOURS AND GALLONS PER MINUTE OF BRINE

Number	Diameter in Inches	Length in Feet	Velocity of Brine through Cooler, Feet per Minute																	
			75						200						400					
			Total brine, gallons per minute	Mean temperature difference of brine and ammonia, degrees F.					Total brine, gallons per minute	Mean temperature difference of brine and ammonia, degrees F.					Total brine, gallons per minute	Mean temperature difference of brine and ammonia, degrees F.				
				10	12½	15	17½	20		7½	10	12½	15	17½		5	7½	10	12½	15
1	26	6	55	5.7	7.18	8.6	10.0	11.5	140	7.74	10.3	12.9	15.5	18.1	290	7.15	10.7	14.3	17.9	21.2
2	26	9	55	8.5	10.7	12.8	14.9	17.2	140	11.5	15.4	19.3	23.2	26.0	290	10.7	16.0	21.4	26.8	32.0
3	26	12	55	11.4	14.4	17.2	20.0	22.9	140	15.5	20.6	25.8	31.0	36.2	290	14.3	21.4	28.6	35.8	42.9
4	34	9	90	13.6	17.2	20.5	23.8	27.4	230	18.5	24.6	30.7	37.0	43.1	440	17.1	25.5	34.1	42.6	51.1
5	34	12	90	18.1	22.8	27.3	31.7	36.4	230	24.6	32.7	41.0	49.2	57.4	440	22.6	33.9	45.3	56.7	68.0
6	34	18	90	27.2	34.3	41.0	47.7	54.8	230	37.0	50.0	62.0	74.0	86.2	440	34.1	51.0	68.2	85.4	101.0
7	42	12	190	33.1	41.8	50.0	58.0	66.5	510	45.0	59.8	74.8	90.0	105.0	970	41.5	62.0	83.0	103.0	123.0
8	42	18	190	50.0	62.6	75.0	87.0	100.0	510	67.4	89.8	112.0	135.0	157.0	970	62.2	93.0	124.0	156.0	187.0
			Velocity of brine through cooler, feet per minute																	
			150						300						500					
1	26	6	110	8.85	11.0	13.2	15.4	17.7	210	9.7	13.0	16.3	19.5	22.7	365	8.0	12.0	16.0	20.0	24.0
2	26	9	110	13.2	16.5	20.0	23.0	27.0	210	14.5	20.0	24.0	29.0	34.0	365	12.0	18.0	24.0	30.0	36.0
3	26	12	110	17.7	22.0	27.0	31.0	36.0	210	20.0	26.0	33.0	39.0	46.0	365	16.0	24.0	32.0	41.0	48.0
4	34	9	180	21.1	28.0	32.0	37.0	42.0	345	23.0	31.0	39.0	47.0	54.0	550	19.0	29.0	39.0	48.0	57.0
5	34	12	180	28.0	35.0	42.0	49.0	56.0	345	31.0	41.0	52.0	62.0	72.0	550	26.0	38.0	51.0	64.0	75.0
6	34	18	180	43.0	53.0	63.0	74.0	85.0	345	47.0	62.0	78.0	93.0	108.0	550	38.0	58.0	77.7	96.0	113.0
7	42	12	380	52.0	64.0	77.0	90.0	102.0	765	57.0	75.0	95.0	113.0	132.0	1200	47.0	70.0	93.0	116.0	138.0
8	42	18	380	77.0	96.0	116.0	134.0	154.0	765	85.0	113.0	141.0	170 0	198.0	1200	70.0	105.0	140.0	175.0	206.0

TABLE 68

Capacities of Double-Pipe Brine Coolers, $1\frac{1}{4}$-In. and 2-In. Pipe, 10 Ft. and 20 Ft. Long Over All, Specifications, Weights

Pipes High	Effective Surface	Effective Surface When Submerged	Height over Top Pipe, Inches	Height over All, Inches	Full Weight Pipe, Weight	Extra Heavy Pipe, Weight	Add for Soldering Joints, Dollars
10-ft. coils							
4	15.0	19.2	24	33	550	610	12
6	19.5	27.8	32	39	750	840	17
8	26.0	38.2	40	47	1000	1120	22
10	32.5	48.0	48	63	1250	1400	28
12	39.0	57.6	56	71	1500	1680	33
14	45.5	67.2	64	79	1750	1960	39
16	51.5	77.0	72	87	2000	2220	44
18	58.2	86.5	80	95	2250	2500	49
20	64.9	96.0	88	113	2500	2800	54
20-ft. coils							
4	30.4	44.4	24	33	720	900	11
6	45.6	66.6	32	39	1080	1320	17
8	61.0	89.0	40	47	1440	1760	22
10	76.5	111.0	48	63	1800	2200	28
12	91.0	133.2	56	71	2130	2620	33
14	106.5	155.5	64	79	2500	3000	37

Gas and liquid valves included with each coil. No headers included.

TABLE 69

Capacities of Double-Pipe Brine Coolers, 2-In. and 3-In. Pipe, 10 Ft. and 20-Ft. Long Over All, Specifications, Weights

Pipes High	Effective Surface	Effective Surface When Submerged	Height over Top Pipe, Inches	Height over All, Inches	Full Weight Pipe, Weight	Extra Heavy Pipe, Weight	Add for Soldering Joints, Dollars
10-ft. coils							
2	10.5	14.4	12	20	430	570	7
4	21.0	28.8	24	32	820	980	12
6	31.5	43.2	36	44	1250	1450	17
8	42.0	57.5	48	56	1650	1930	22
10	52.5	72.0	60	68	2000	2400	28
12	63.0	86.5	72	80	2400	2880	33
14	73.5	101.0	84	92	2820	3360	39
16	84.0	115.0	96	104	3220	3840	44
18	94.2	129.0	108	116	3620	4320	49
20	105.0	144.0	120	128	4050	4800	54
20-ft. coils							
2	23.0	31.5	12	20	680	845	7
4	46.0	63.0	24	32	1320	1660	12
6	69.0	94.5	36	44	1960	2460	17
8	92.0	126.0	48	56	2600	3300	22
10	115.0	157.6	60	68	3240	4100	28
12	138.0	189.1	72	80	3880	4950	33
14	161.0	220.5	84	92	4560	5760	39
16	184.0	252.0	96	104	5200	6600	44
18	207.0	283.6	108	116	5840	7400	49
20	230.0	315.2	120	128	6480	8200	54

Gas and liquid valves included with each coil. Add for gas and liquid headers 40 lb. per coil; for brine valves and headers 75 lb. per coil.

TABLE 70

CAPACITIES OF MULTIPASS SHELL AND TUBE BRINE COOLERS, SPECIFICATIONS, WEIGHTS

Number	Diameter, Inside, in Inches	Length over Tube Heads, Inches	Number of 2-In. Tubes	Number of Brine Passes	Thickness of Shell, Inches	Thickness of Heads, Inches	Diameter of Brine Connections, Inches	Effective Tube Surface	Weight, Pounds
1	26	6	60	12	$\frac{1}{4}$	$\frac{7}{8}$	$3\frac{1}{2}$	172	2,750
2	26	9	60	12	$\frac{1}{4}$	$\frac{7}{8}$	$3\frac{1}{2}$	257	3,400
3	26	12	60	12	$\frac{1}{4}$	$\frac{7}{8}$	$3\frac{1}{2}$	344	4,050
4	34	9	96	12	$\frac{5}{16}$	1	4	410	5,900
5	34	12	96	12	$\frac{5}{16}$	1	4	545	7,275
6	34	18	96	12	$\frac{5}{16}$	1	4	820	8,650
7	42	12	176	10	$\frac{3}{8}$	1	6	998	10,800
8	42	18	176	10	$\frac{3}{8}$	1	6	1498	14,250

Head in feet and corresponding pounds per square inch for given brine velocities (Feet per minute)

	75		150		200		300		400		500	
	Ft.	Lb.	Ft.	Lb.	Ft.	Lb.	Ft.	Lb.	Ft.	Lb.	Ft.	Lb.
1	.56	.24	2.1	9.1	3.5	1.5	7.5	3.3	13.0	5.7	20.0	8.7
2	.75	.32	2.8	1.2	4.6	2.0	9.8	4.3	17.0	7.4	26.0	12.0
3	1.0	.43	3.5	1.5	5.7	2.5	12.0	5.2	20.0	8.7	33.0	15.0
4	.75	.32	2.8	1.2	4.6	2.0	9.8	4.3	17.0	7.4	26.0	12.0
5	1.0	.43	3.5	1.5	5.7	2.5	12.0	5.2	20.0	8.7	33.0	15.0
6	1.5	.65	4.8	2.1	8.0	3.5	17.0	7.4	28.0	12.1	45.0	20.0
7	1.0	.43	3.3	1.4	5.6	2.4	11.0	4.8	19.0	8.2	32.0	14.0
8	1.4	.61	4.6	2.0	8.0	3.5	16.0	7.0	27.0	12.0	44.0	19.0

TO USE CAPACITY TABLE

Given: Temperature of room or substance to be cooled = 32 degrees
Tons refrigeration required = 50

Assume: Standard Ammonia Suction pressure of 16 lb., corresponding to 0 deg. F. From 5 deg. to 10 deg. temperature difference should be assumed between ammonia and brine leaving cooler and between brine entering cooler and temperature of room. Using 10 deg. for this case gives:

Temperature of brine entering cooler = 22 degrees
Temperature of brine leaving cooler = 10

$$\text{Mean temperature difference of brine and ammonia} = \frac{22 + 10}{2} - 0 = 16 \text{ deg.}$$

NOTE—Weights include Ammonia Safety Relief valve, Liquid Gage, Pressure Gage, Gas and Liquid Valves and C. I. supports—no Brine Valves included. Cooler Shells are made of flange steel and Tubes are genuine Charcoal Iron Boiler Tubes.

From Table: Next select a brine velocity which will be most suitable from all viewpoints, such as first cost of cooler and brine pump, and power required for pumping brine. Assuming 400 ft. per minute brine velocity through cooler; for 50 tons refrigeration under the conditions stated one finds in the table under 15 degrees mean temperature difference that a 34-in. by 9-ft. cooler has a capacity of 51 tons of refrigeration. The head in feet and corresponding pounds per square inch from table above = 17 ft. or 7.4 lb. per square inch to overcome friction or resistance to flow through cooler.

TABLE 71

Sodium Chloride—NaCl

Specific Heat * (Temperature Table)

Jessup—Properties of Refrigerating Brines. Refrigerating Engineering, December, 1925. By permission of Director of Bureau of Standards

Degrees F.	Degrees C	Specific Heats of Various Per Cent NaCl Solutions at Various Temperatures																				
− 4	−20	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	0.784		
+14	−10	...	....	....	....	....	....	....	....	....	....	0.835	0.827	0.821	0.815	0.809	0.804	0.798	0.794	.789	0.784	0.779
32	0	0.936	0.924	0.913	0.902	0.892	0.882	0.873	0.865	0.836	0.848	.841	.834	.827	.821	.815	.809	.803	.798	.793	.788	.783
50	+10	.938	.927	.916	.906	.896	.887	.878	.869	.861	.853	.846	.839	.832	.825	.819	.813	.807	.801	.796	.791	.786
68	20	.940	.929	.919	.909	.900	.890	.882	.873	.865	.857	.849	.842	.835	.828	.822	.815	.809	.803	.798	.792	.787
86	30	.942	.932	.922	.912	.902	.893	.884	.876	.868	.859	.852	.844	.837	.830	.823	.817	.811	.804	.799	.793	.788
Per Cent of NaCl....		5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
Specific gravity †..		1.036	1.044	1.051	1.058	1.066	1.073	1.081	1.088	1.096	1.104	1.112	1.119	1.127	1.135	1.143	1.151	1.159	1.167	1.176	1.184	1.192
Degrees Baume.. .		5.08	6.06	7.04	8.00	8.96	9.91	10.85	11.78	12.71	13.63	14.54	15.45	16.35	17.25	18.14	19.02	19.90	20.78	21.65	22.51	23.37
Degrees F., Freezing Pt., Degrees F....		26.7	25.5	24.2	22.9	21.6	20.2	18.8	17.3	15.7	14.1	12.4	10.6	8.7	6.7	4.6	2.4	0.0	−2.5	−5.2	+1.4	+13.3

Per Cent of Sodium Chloride and Corresponding Freezing Points (Specific Gravity Table)

Specific gravity †..	1.03	1.04	1.05	1.06	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20
Per cent NaCl.......	4.14	5.51	6.86	8.21	9.56	10.88	12.20	13.51	14.81	16.09	17.36	18.63	19.87	21.11	22.33	23.54	24.74	25.93
Degrees F., Freezing Pt., Degrees F......	27.6	26.0	24.4	22.7	20.8	19.0	17.0	14.9	12.7	10.4	8.0	5.4	2.7	−0.3	3.5	−3.0	+9.0	+27.0

Per Cent of Sodium Chloride and Corresponding Freezing Points (Baumé Density Table)

Degrees Baumé	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
Per cent NaCl...	4.92	5.94	6.96	8.00	9.04	10.10	11.16	12.23	13.31	14.40	15.50	16.61	17.72	18.84	19.97	21.11	22.26	23.41	24.57
Degrees F.—F.P	26.7	25.5	24.3	22.9	21.6	20.1	18.6	16.9	15.2	13.4	11.5	9.4	7.3	5.0	2.4	−0.3	−3.2	−5.0	+8.0

* Specific heat expressed in 20 deg. Cal-g deg. C.

† Specific gravity, based on 60 deg. F. water and 60 deg. F. brine; Degrees Be, Beaumé density; degrees F.—F. P., Freezing point.

TABLE 72

Calcium Chloride—$CaCl_2$ (Pure)

Specific Heat * (Temperature Table)

Jessup—Properties of Refrigerating Brines. Refrigerating Engineering, December, 1925.

Degrees F.	Degrees C.	Specific Heats of Various Per Cent Calcium Chloride Solutions at Various Temperatures																						
−40	−40	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....			0.643	0.636	0.635	0.631
−22	−30	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.663	0.656	.649	.642	.639	.635
− 4	−20	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.704	0.695	0.686	0.678	.670	.663	.656	.649	.644	.638
14	−10	.....	.....	.....	.....	.....	.....	.....	0.781	0.768	0.756	0.745	0.734	0.723	.713	.704	.694	.686	.678	.670	.663	.656	.649	.643
32	0	0.882	0.867	0.853	0.839	0.825	0.812	0.799	.787	.775	.763	.752	.741	.731	.721	.711	.702	.693	.685	.677	.669	.662	.655	.648
50	10	0.887	0.872	0.858	0.844	0.831	0.818	0.805	0.793	0.781	0.770	0.759	0.748	0.738	0.728	0.718	0.709	0.7[illegible]	0.692	0.683	0.676	0.669	0.662	0.655
68	20	.892	.877	.863	.849	.836	.823	.811	.799	.787	.775	.764	.754	.744	.733	.724	.715	.706	.698	.690	.682	.675	.668	.661
86	30	.897	.882	.868	.855	.842	.829	.817	.805	.792	.780	.769	.759	.749	.739	.729	.721	.712	.704	.696	.689	.682	.675	.668
Per cent of $CaCl_2$		8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Specific gravity †		1.069	1.078	1.087	1.096	1.105	1.114	1.124	1.133	1.143	1.152	1.162	1.172	1.182	1.192	1.202	1.212	1.223	1.233	1.244	1.254	1.265	1.276	1.287
Degrees Baumé		9.33	10.45	11.57	12.68	13.78	14.88	15.96	17.04	18.12	19.18	20.24	21.28	22.32	23.35	24.38	25.38	26.41	27.41	28.41	29.41	30.39	31.37	32.34
Freezing point		24.2	22.8	21.4	19.8	18.2	16.3	14.4	12.2	9.9	7.4	4.7	1.9	−1.0	−4.0	−7.3	−10.6	−14.1	−18.0	−22.0	−27.0	−32.0	−39.0	−46.0

Per Cent of Calcium Chloride and Corresponding Freezing Points (Specific Gravity Table) *(Jessup)*

Specific gravity †	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20	1.21	1.22	1.23	1.24	1.25	1.26	1.27	1.28	1.29
Per cent of $CaCl_2$	8.14	9.26	10.36	11.45	12.53	13.60	14.66	15.71	16.75	17.78	18.80	19.81	20.81	21.79	22.78	23.74	22.70	25.65	26.59	27.52	28.44	29.35	30.26
Freezing point, degrees F.	24.0	22.5	20.8	19.1	17.2	15.1	30.0	10.6	8.1	5.4	2.6	−0.4	−3.4	−6.4	−9.7	−13.2	−16.8	−20.7	−25.0	−30.0	−35.0	−41.0	−48.0

Per Cent of Calcium Chloride and Corresponding Freezing Points (Baumé Density Table)

Degrees Baumé	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32
Per cent of $CaCl_2$	8.59	9.49	10.39	11.29	12.20	13.11	14.04	14.96	15.89	16.83	17.78	18.73	19.69	20.66	21.63	22.62	23.60	24.59	25.59	26.59	27.61	28.62	29.64
Freezing point, degrees F.	23.4	22.1	20.8	19.3	17.8	16.1	14.3	12.3	10.1	7.8	5.3	2.7	−0.1	−3.0	−6.1	−9.3	−12.7	−16.4	−20.4	−25.0	−30.0	−36.0	−43.0

TABLE 73

Heat Content, B.t.u. per Pound from 32 Deg. F.

E. F. Mueller, A. S. R. E. Journal, July, 1919

Degrees F.	Specific Gravity 1.18	1.20	1.22	1.24	1.26	1.28
−20	......	......	......	−34.82	−34.09	−33.41
−19	......	......	......	−34.16	−33.42	−32.78
−18	......	......	−34.28	−33.50	−32.77	−32.14
−17	......	......	−33.61	−32.84	−32.13	−31.51
−16	......	......	−32.93	−32.18	−31.48	−30.87
−15	......	......	−32.25	−31.52	−30.83	−30.24
−14	......	......	−31.58	−30.86	−30.18	−29.60
−13	......	......	−30.90	−30.19	−29.63	−28.96
−12	......	......	−30.22	−29.53	−28.88	−28.33
−11	......	......	−29.54	−28.86	−28.23	−27.69
−10	......	......	−28.86	−28.20	−27.59	−27.05
−9	......	......	−28.18	−27.54	−26.94	−26.41
−8	......	−28.21	−27.50	−26.87	−26.28	−25.78
−7	......	−27.51	−26.82	−26.21	−25.63	−25.14
−6	......	−26.82	−26.14	−25.54	−24.98	−24.50
−5	......	−26.12	−25.46	−24.88	−24.33	−23.86
−4	......	−25.42	−24.78	−24.21	−23.68	−23.22
−3	......	−24.72	−24.10	−23.54	−23.03	−22.58
−2	−24.66	−24.02	−23.42	−22.88	−22.38	−21.94
−1	−23.94	−23.32	−22.74	−22.21	−21.72	−21.30
0	−23.22	−22.62	−22.05	−21.54	−21.07	−20.66
+1	−22.50	−21.92	−21.37	−20.88	−20.42	−20.02
2	−21.78	−21.22	−20.69	−20.21	−19.76	−19.38
3	−21.06	−20.52	−20.09	−19.54	−19.11	−18.74
4	−20.34	−19.82	−19.32	−18.87	−18.46	−18.10
5	−19.62	−19.12	−18.63	−18.20	−17.80	−17.46
6	−18.90	−18.41	−17.95	−17.53	−17.15	−16.81
7	−18.18	−17.71	−17.26	−16.86	−16.49	−16.17
8	−17.45	−17.01	−16.58	−16.19	−15.84	−15.53
9	−16.73	−16.36	−15.89	−15.52	−15.18	−14.88

Degrees F.	Specific Gravity 0.999	1.05	1.10	1.15	1.18	1.20	1.22	1.24	1.26	1.28
10	......	......	......	−16.70	−16.01	−15.60	−15.20	−14.85	−14.52	−14.24
11	......	......	......	−15.95	−15.28	−14.89	−14.52	−14.18	−13.87	−13.60
12	......	......	......	−15.19	−14.56	−14.19	−13.83	−13.51	−13.21	−12.95
13	......	......	......	−14.44	−13.84	−13.48	−13.14	−12.83	−12.55	−12.31
14	......	......	......	−13.68	−13.11	−12.78	−12.45	−12.16	−11.90	−11.66
15	......	......	......	−12.93	−12.39	−12.07	−11.76	−11.49	−11.24	−11.02
16	......	......	......	−12.17	−11.66	−11.36	−11.08	−10.82	−10.58	−10.37
17	......	......	......	−11.41	−10.94	−10.66	−10.38	−10.14	−9.99	−9.73
18	......	......	......	−10.65	−10.21	−9.95	−9.70	−9.47	−9.26	−9.08
19	......	......	......	−9.89	−9.48	−9.24	−9.01	−8.80	−9.60	−8.93
20	......	......	......	−9.14	−8.76	−8.53	−8.32	−8.12	−7.94	−7.79
21	......	......	−9.11	−8.38	−8.03	−7.80	−7.62	−7.44	−7.28	−7.14
22	......	......	−8.28	−7.62	−7.30	−7.11	−6.93	−6.77	−6.62	−6.49
23	......	......	−7.46	−6.86	−6.57	−6.40	−6.24	−6.09	−5.96	−5.84
24	......	......	−6.33	−6.10	−5.84	−5.69	−6.55	−5.42	−5.30	−5.20
25	......	......	−5.80	−5.34	−5.12	−4.98	−4.86	−4.74	−4.64	−4.55
26	......	......	−4.98	−4.57	−4.38	−4.27	−4.16	−4.07	−3.98	−3.90
27	......	......	−4.15	−3.81	−3.65	−3.50	−3.47	−3.39	−3.32	−3.25
28	......	......	−3.32	−3.05	−2.92	−2.85	−2.78	−2.71	−2.65	−2.60
29	......	−2.74	−2.49	−2.29	−2.19	−2.14	−2.08	−2.03	−1.99	−1.95
30	......	−1.83	−1.66	−1.53	−1.46	−1.43	−1.39	−1.36	−1.33	−1.30
31	......	−0.91	−0.83	−0.76	−0.73	−0.71	−0.70	−0.68	−0.66	−0.65
32	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
35	3.02	+2.74	+2.49	+2.29	+2.20	+2.14	+2.09	+2.04	+1.99	+1.95
40	8.05	7.30	6.65	6.12	5.87	5.72	5.57	5.44	5.32	5.22
35	13.07	11.87	10.82	9.96	9.55	9.30	9.07	8.86	8.66	8.49
50	18.08	16.44	15.00	13.81	13.24	12.90	12.57	12.28	12.01	11.77
55	23.08	21.02	19.18	17.67	16.94	16.50	16.09	15.72	15.37	15.06
60	28.08	25.60	23.38	21.54	20.65	20.12	19.61	19.16	18.73	18.36
65	33.07	30.18	27.58	25.42	24.37	23.74	23.14	22.61	22.11	21.67

Weights

Degrees F.	Specific Gravity																		
	0.999	1.05	1.10	1.15	1.18	1.20	1.22	1.24	1.26	1.28	1.05	1.10	1.15	1.18	1.20	1.22	1.24	1.26	1.28
	Pounds per cubic foot										Pounds per gallon								
−10	......	......	......	......	......	......	77.2	78.5	79.8	81.1	......	......	......	......	......	10.32	10.49	10.66	10.84
0	......	......	......	......	74.4	75.7	77.0	78.3	79.6	80.9	......	......	......	9.95	10.13	10.30	10.47	10.64	10.81
10	......	......	......	72.4	74.3	75.6	76.9	78.2	79.5	80.8	......	......	9.68	9.93	10.11	10.28	10.45	10.62	10.79
20	......	......	69.1	72.3	74.2	75.4	76.7	78.0	79.3	80.6	......	9.23	9.66	9.92	10.09	10.26	10.43	10.60	10.77
30	......	65.8	69.0	72.1	74.0	75.3	76.6	77.9	79.2	80.4	8.79	9.22	9.64	9.90	10.07	10.24	10.41	10.58	10.75
40	62.4	65.7	68.9	72.0	73.9	75.2	76.4	77.7	79.0	80.3	8.78	9.21	9.63	9.88	10.05	10.22	10.39	10.56	10.73
50	62.4	65.6	68.8	71.9	73.8	75.0	76.3	77.6	78.8	80.1	8.77	9.19	9.61	9.86	10.03	10.20	10.37	10.54	10.70
60	62.4	65.6	68.7	71.8	73.7	74.9	76.2	77.4	78.7	79.9	8.76	9.18	9.60	9.85	10.01	10.18	10.35	10.52	10.68
70	62.3	65.5	68.6	71.7	73.5	74.8	76.0	77.3	78.5	79.7	8.75	9.17	9.58	9.83	10.00	10.16	10.33	10.49	10.66

Specific gravity (60 deg. F.–39 F.). Mass of Unit Volume of 60 deg. Brine. Mass of Unit Volume of 39 deg. Water.

TABLE 74

Magnesium Chloride—$MgCl_2$

Specific Heat* (Temperature Table)

Jessup—Properties of Refrigerating Brines. Refrigerating Engineering, December, 1925

Degrees F.	Degrees C.	Specific Heats of Various Per Cent $MgCl_2$ Solutions at Various Temperatures																									
−22	−30	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	0.711	0.700	0.690								
−4	−20	....	....	....	....	....	....	....	....	....	....	....	....	....	0.739	0.728	.717	.804	.696	0.686	0.676	0.666	0.657				
14	−10	....	....	....	....	....	....	....	0.822	0.809	0.795	0.782	0.770	0.758	.746	.735	.724	.713	.702	.692	.682	.673	.664	0.654	0.654	0.637	0.629
32	0	0.927	0.912	0.897	0.882	0.868	0.854	0.840	.827	.814	.801	.788	.776	.764	.752	.741	.730	.719	.709	.698	.689	.679	.670	.669	.652	.613	.635
50	10	.929	.914	.900	.885	.872	.858	.845	.832	.819	.806	.794	.782	.770	.758	.747	.737	.726	.715	.705	.695	.685	.676	.666	.658	.649	.641
68	20	0.931	0.917	0.903	0.889	0.875	0.862	0.849	0.837	0.824	0.812	0.800	0.788	0.776	0.765	0.754	0.743	0.732	0.722	0.711	0.702	0.692	0.683	0.673	0.664	0.655	0.647
86	30	.933	.919	.905	.892	.879	.866	.854	.841	.830	.817	.805	.794	.783	.771	.760	.750	.739	.728	.718	.708	.698	.689	.679	.671	.661	.653
Per cent $MgCl_2$		5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Specific gravity †		1.043	1.051	1.060	1.069	1.078	1.086	1.096	1.105	1.114	1.123	1.132	1.142	1.151	1.161	1.170	1.180	1.190	1.200	1.210	1.220	1.230	1.241	1.251	1.262	1.273	1.283
Degrees Bé		5.91	7.06	8.20	9.32	10.44	11.53	12.64	13.73	14.82	15.88	16.94	17.99	19.03	20.08	21.11	22.13	23.14	24.17	25.16	26.16	27.15	28.14	29.12	30.09	31.06	32.02
Degrees F.—F.P.		26.4	25.0	23.5	21.8	20.0	17.9	15.7	13.1	10.3	7.3	4.0	0.4	−3.5	−7.7	−12.2	−17.3	−23.0	−27.0	−20.0	−14.0	−10.0	−6.0	−3.0	−1.0	+1.0	+2.0

Per Cent of Magnesium Chloride and Corresponding Freezing Points (Specific Gravity Table) *(Jessup)*

Specific gravity †	1.04	1.05	1.06	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20	1.21	1.22	1.23	1.24	1.25	1.26	1.27	1.28
Per cent of $MgCl_2$	4.71	5.86	7.01	8.15	9.27	10.40	11.50	12.59	13.67	14.75	15.83	16.88	17.93	18.96	19.99	21.01	22.00	23.00	23.99	29.96	25.92	26.88	27.82	28.76	29.68
Degrees F.—F. P.	26.7	25.2	23.5	21.5	29.4	17.0	14.4	11.5	8.3	4.8	1.0	−3.0	−7.4	−12.0	−17.2	−23.1	−27.0	−20.0	−14.0	−10.0	−6.0	−3.0	−1.0	+1.0	+2.0

Per Cent of Magnesium Chloride and Corresponding Freezing Points (Baumé Density Table)

Degrees Bé	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32
Per cent $MgCl_2$	5.94	6.82	7.72	8.61	9.51	10.42	11.32	12.25	13.17	14.11	15.06	16.01	16.97	17.93	18.89	19.87	20.86	21.84	22.83	23.83	24.85	25.86	26.88	27.90	28.94	29.98
Degrees F.—F.P.	25.1	23.8	22.3	20.8	19.0	17.0	14.9	12.5	9.9	7.0	3.8	0.3	−3.3	−7.3	−11.6	−16.4	−22.0	−28.0	−21.0	−15.0	−10.0	−6.0	−3.0	−1.0	+1.0	+2.0

* Specific heat, expressed in **68 deg. per B.t.u. pound, Deg. F. (20 deg. Cal per g. deg. C.) and in 59 deg. B.t.u. per pound, deg. F. (15 deg. Cal per g., deg. C.).**

† Specific gravity, based on 60 deg. F. water and 60 deg. F. brine; degrees Bé, Baumé density; degrees F.—F.P., Freezing point.

CHAPTER IX

THE WATER SUPPLY

Heat removed by the Condenser.—With the exception of the household refrigerating machine, which is air cooled in the majority of cases, all refrigerating machines use a water cooled condenser in order to absorb the heat which must be removed at the upper temperature in order to liquefy the refrigerant again. The quantity of heat requiring to be removed theoretically is equal to the refrigerating effect plus the work done on the refrigerant by the compressor, provided an expansion cylinder instead of the pressure reducing valve is used. In practice this statement is not correct, because of the use of the irreversible throttling process and because of the heating effect of the compressor which means that more work is done on the gas than would be necessary with non-condensing cylinder walls. As dry compression is used in America almost entirely, the temperature of discharge from the compressor is frequently quite high, as high as 300 deg. F. with certain operating pressures. If the condenser is on the roof, a large amount of this superheat is lost to the atmosphere before the gas reaches the condenser, whereas self-contained units and particularly the shell and tube design of condenser with a short run to the compressor will lose very little superheat before the gas reaches the condenser. The result is that the condensing water has to absorb an amount of heat varying from 220 to 300 B.t.u. per ton of refrigeration per minute.

The amount of water to be used in the plant varies with operating and economic conditions. As already noted (Chapter II), the condenser pressure, the pressure corresponding to the temperature of liquefaction, affects the horse power per ton of refrigeration and also the capacity of the compressor. It is therefore desirable to reduce the condenser pressure as much as it is compatible with economy of operation, whence it is obvious that one may either use a large amount of water that will be raised but a few degrees in temperature or employ a small amount of water to be raised in temperature a relatively large number of degrees. Referring to Table 75,[1] the theoretical values of the amount of water required are

[1] The Electrically Operated Ice Plant, Sloan, Journal of Am. Soc. Refrigerating Engineers.

TABLE 75

GALLONS OF CONDENSER WATER PER TON OF REFRIGERATION

Condenser Pressure, Pounds per Square Inch, Gage	Corresponding Temperature, Degrees F.	60 Deg. F. Water				70 Deg. F. Water				80 Deg. F. Water				90 Deg. F. Water			
		Range, degrees F.	Water per ton of refrigeration, gallons per minute of suction pressure, pounds gage			Range, degrees F.	Water per ton of refrigeration, gallons per minute of suction pressure, pounds gage			Range, degrees F.	Water per ton of refrigeration, gallons per minute of suction pressure, pounds gage			Range, degrees F.	Water per ton of refrigeration, gallons per minute of suction pressure, pounds gage		
			15	20	25		15	20	25		15	20	25		15	20	25
126	75	10	2.90	2.85	2.80												
131	77	12	2.40	2.35	2.30												
136	79	14	2.05	2.00	1.95												
141	81	16	1.85	1.80	1.75	6	4.87	4.80	4.70								
147	83	18	1.65	1.60	1.55	8	3.67	3.60	3.55								
152	85	20	1.47	1.45	1.41	10	2.95	2.90	2.85								
158	87	22	1.35	1.32	1.30	12	2.45	2.40	2.35								
163	89	24	1.25	1.22	1.20	14	2.10	2.05	2.00								
169	91	26	1.15	1.13	1.10	16	1.90	1.85	1.80	6	5.00	4.90	4.80				
175	93	28	1.08	1.05	1.03	18	1.68	1.63	1.60	8	3.75	3.70	3.65				
181	95	30	1.00	0.99	0.97	20	1.50	1.48	1.45	10	3.00	2.95	2.90				
187	97	32	0.95	0.93	0.90	22	1.38	1.35	1.32	12	2.52	2.48	2.43				
194	99	......	......	......	......	24	1.28	1.25	1.22	14	2.13	2.10	2.06				
200	101	......	......	......	......	26	1.17	1.15	1.13	16	1.92	1.85	1.83	6	5.10	5.00	4.90
207	103	......	......	......	......	28	1.10	1.07	1.05	18	1.70	1.67	1.63	8	3.83	3.75	3.70
214	105	......	......	......	......	30	1.02	1.00	0.98	20	1.53	1.50	1.48	10	3.07	3.00	2.95
221	107	......	......	......	......	32	0.95	0.93	0.92	22	1.40	1.37	1.35	12	2.57	2.51	2.47
229	109	......	......	......	......	......	......	......	......	24	1.30	1.28	1.25	14	2.19	2.13	2.10
236	111	......	......	......	......	......	......	......	......	26	1.20	1.16	1.15	16	1.95	1.90	1.88
244	113	......	......	......	......	......	......	......	......	28	1.12	1.08	1.06	18	1.74	1.69	1.65
251	115	......	......	......	......	......	......	......	......	30	1.05	1.02	1.00	20	1.56	1.53	1.50
259	117	......	......	......	......	......	......	......	......	32	1.00	0.98	0.95	22	1.43	1.40	1.37

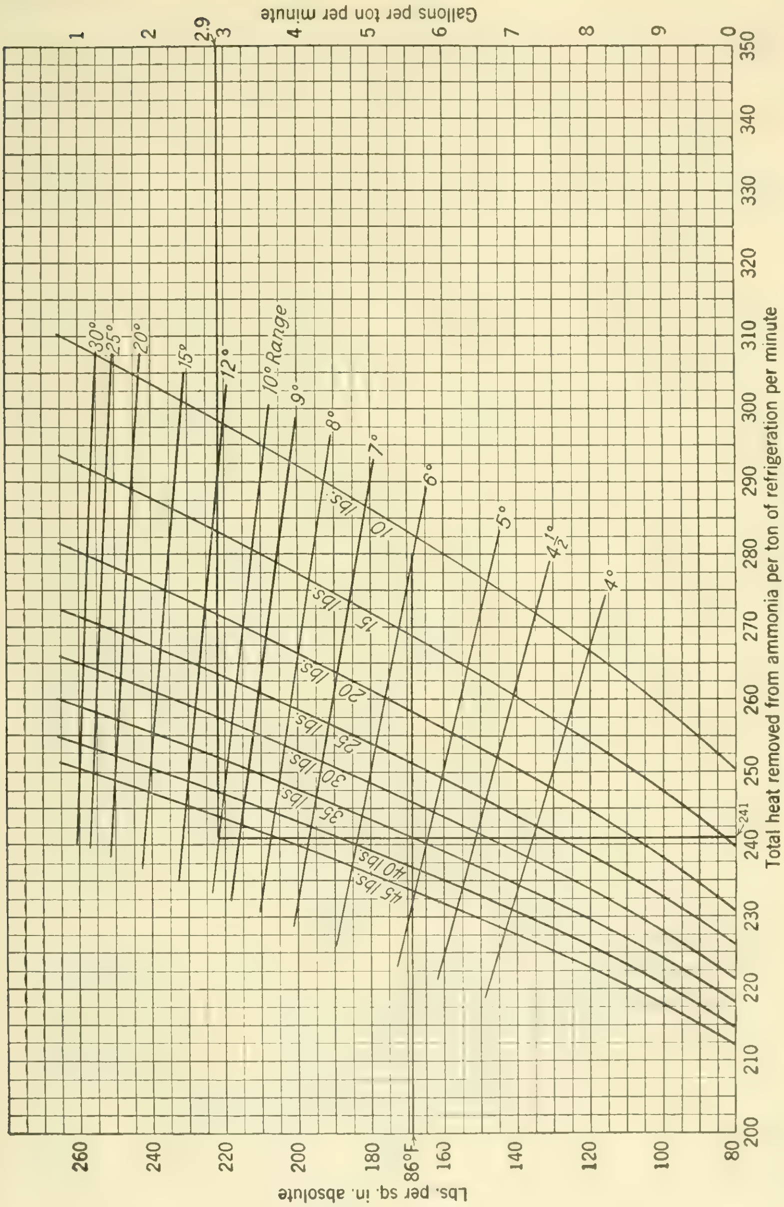


FIG. 182.—Condensing Water Chart.

listed for various initial temperatures and number of degrees rise of temperature of the water. Whether deep well, cooling tower or spray nozzle water is used a certain amount of pumping is required, and the amount of this pumping is justified differently in each type of installation. This point is well brought out by a chart of the heat absorbed by the condenser water, as shown in Fig. 182.

In Fig. 182 the ordinates are condenser pressures, and the abscissa are the heat removed (assuming adiabatic compression and dry saturated gas entering the compressor) per ton of refrigeration per minute. The number of gallons of condensing water can be found also from the scale on the right-hand side of the chart. As an example of the use of the diagram take standard conditions of operation, which are 86 degree condenser and 5 degree evaporator temperatures (corresponding to 154.5 lb. and 19.6 lb. gage, respectively, for ammonia) from which the heat to be removed is found to be 241 B.t.u. per ton per minute. Taking a ten-degree rise of water temperature this condition can be obtained, as regards condenser and suction pressure, with the use of 2.9 gal. of water per minute at an initial temperature of 70 deg. Assuming instead a 5-degree rise and 5.8 gal. per ton of refrigeration per minute, the new condenser pressure will be (still assuming that there is no air or other non-condensible gas present) that corresponding to $70 + 5 + 5 = 80$ deg. F., or 138.3 lb. gage. Therefore, doubling the amount of water has had the effect of reducing the head pressure 16.2 lb. and the horse power per ton of refrigeration (Fig. 182) from about 1.29 to about 1.15. In some cases the use of a large amount of water will be found to be justified, especially if the total cost of pumping, plus the overhead due to the piping and machinery, the sprays or the cooling tower, if such have to be used, is not very large. As a rule when the condenser pressure gets over 175 lb. gage, it is wise to increase the volume of water in order to improve in other ways the operating conditions in the compressor room.

Temperature of Well and Surface Water.—The most satisfactory source of condensing water is that of the deep well. Figure 183, taken from the United States Geological Survey statistics,[2] shows the approximate temperature of the water from non-thermal wells at a depth of 30 to 60 feet, and Fig. 184 gives the corresponding temperature of the surface water. In this bulletin it is stated: "The temperature of the water in the ground at any place is in general about the same as the mean *annual temperature* of the *air*. Near the surface the temperature of the water follows the temperature changes of the air; at greater

[2] Water Survey Paper 520-F, by W. D. Collins.

depths the water temperature corresponds to the increase in the earth temperature with increasing depth." Table 76 also gives the approxi-

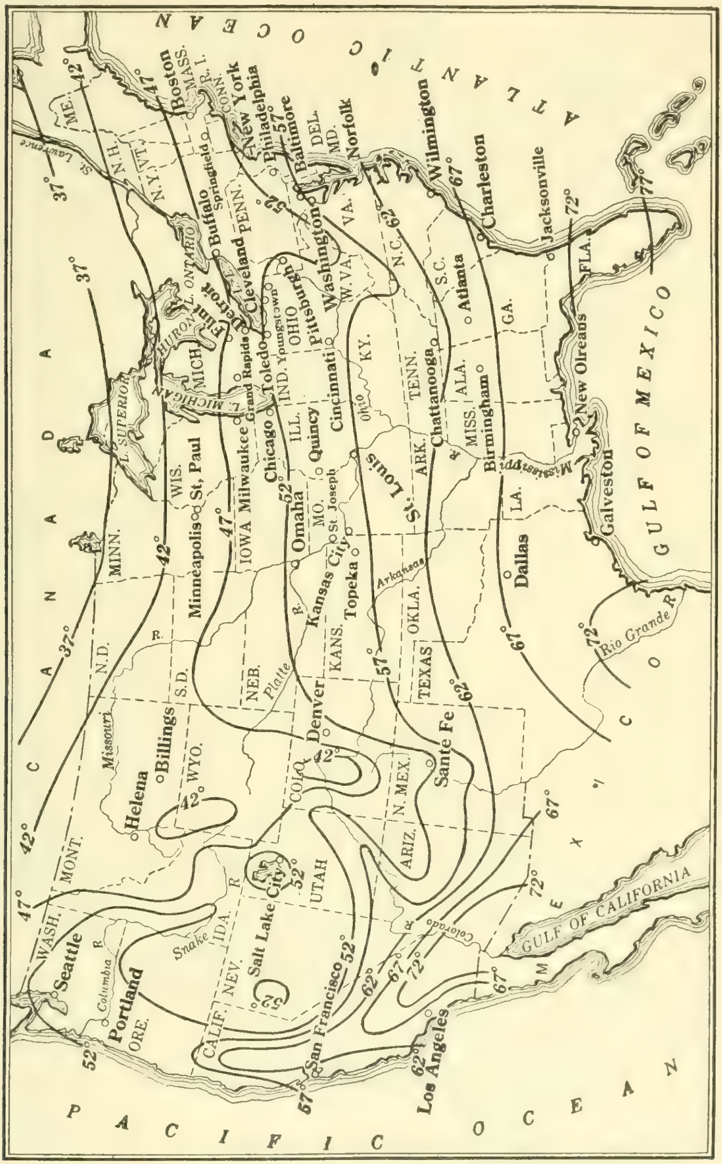


FIG. 183.—Temperature of Non-Thermal Wells.

mate temperature of the surface water and of the air. Very naturally these should be closely related.

Types of Pumps for Deep Wells.—There are three methods of pumping deep wells: the reciprocating pump, the Pohlé air lift (Fig. 185), and

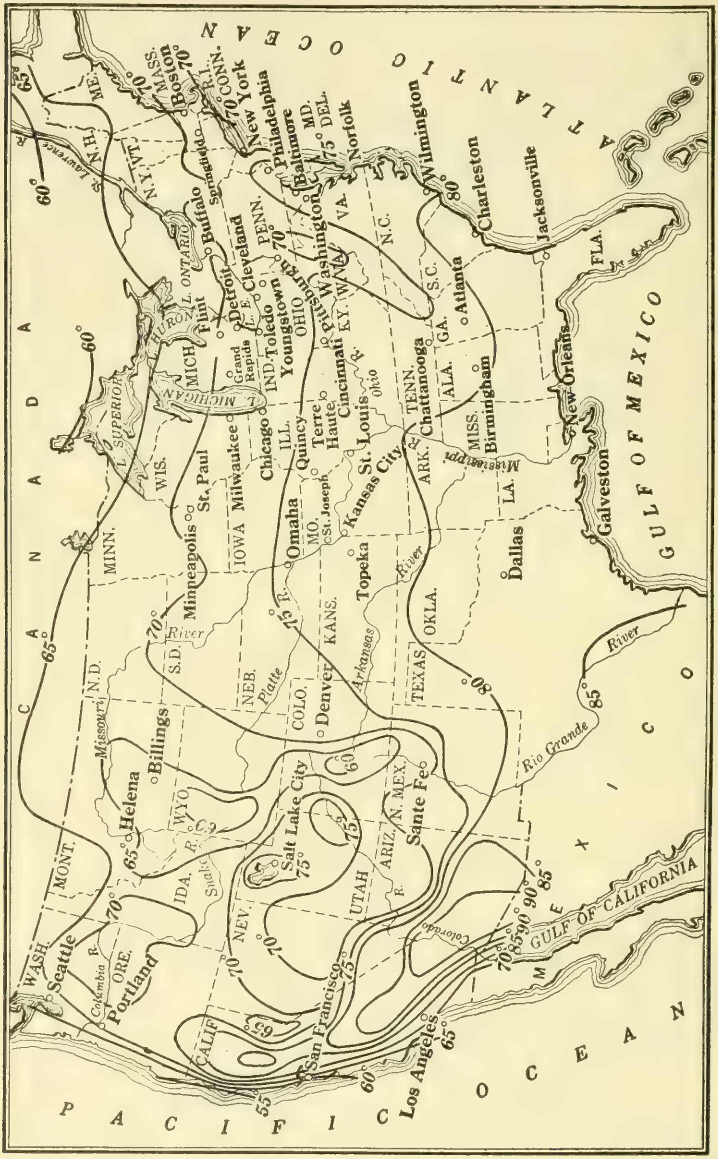


Fig. 184.—Temperature of Surface Water During Summer Months.

the centrifugal pump. The reciprocating pump is best suited to small quantities of water and is usually operated at the high efficiency of

67 to 73 per cent.[3] There is likelihood of trouble, however, due to leaky valves, and the capacity of the well is usually limited to 400 to

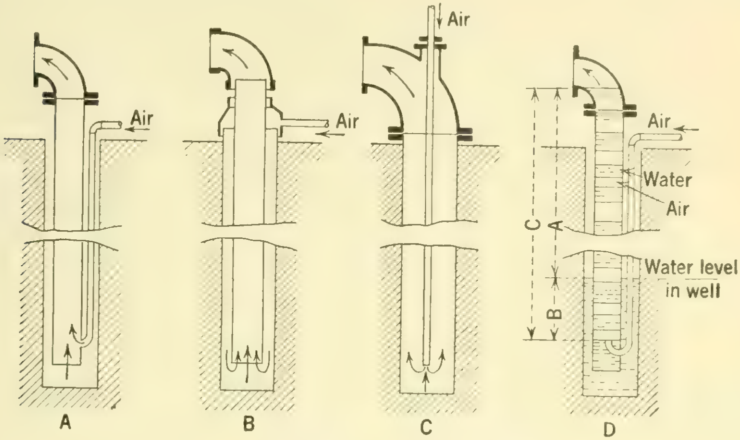


FIG. 185.—The Pohlé Air Lift.

600 gal. per min. so that if larger quantities are required more wells must be sunk.

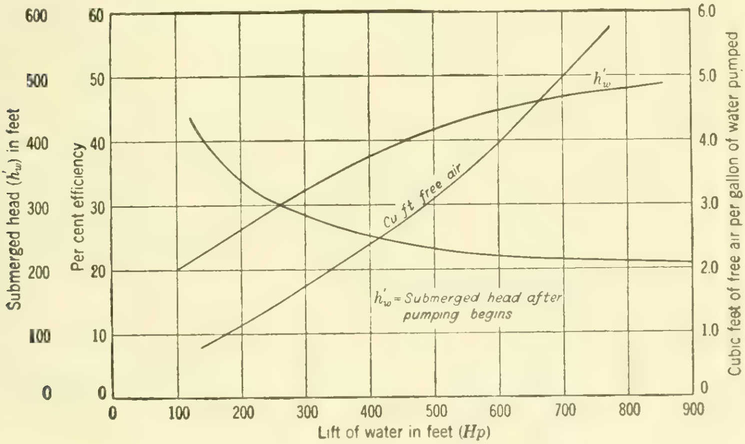


FIG. 186.—Performance of Air Lift Pumps.

The air lift, first proposed by Dr. Julius Pohlé in 1888, can raise much more water from the same sized well than can the plunger pump. Armour & Co., Chicago, were able in 1912 to increase their quantity of

[3] Deep Well Water Pumping, G. B. Mulley, Power, 6–23–25.

TABLE 76

Mean Monthly Temperature of Surface Water and Air

W = Water temperatures from waterworks officials.
A = Air temperatures from U. S. Weather Bureau.

Locality	Year	Mean Monthly Temperature																								Maximum Daily Temperature of Water	
		January		February		March		April		May		June		July		August		September		October		November		December			
		W	A	W	A	W	A	W	A	W	A	W	A	W	A	W	A	W	A	W	A	W	A	W	A	July	Aug.
1. Baltimore, Md.	1918	35	24	34	35	43	47	50	53	65	69	69	71	71	75	74	78	67	65	57	61	50	47	40	42	76	78
	1919	37	38	37	38	45	46	50	53	57	65	67	73	73	78	72	75	68	70	61	63	48	47	39	33	77	76
	1920	35	29	34	33	39	45	51	53	59	61	69	73	74	76	75	75	70	70	62	63	48	47	40	40	77	78
	1921	36	37	39	39	51	55	58	59	61	63	73	75	76	80	73	74	72	74	58	58	46	48	37	37	80	80
	1922	34	33	36	38	47	45	60	56	69	67	73	75	76	77	75	74	71	71	62	61	49	49	38	37	78	77
	1923	34	37	35	32	39	44	48	54	57	64	64	77	66	77	69	75	66	70	61	57	52	46	44	45	68	73
2. Birmingham, Ala.	1923	48	52	47	46	51	54	58	62	67	69	72	77	80	78	81	78	78	76	69	64	63	52	53	53		
3. Chattanooga, Tenn.	1923	49	47	49	42	52	51	58	59	65	65	73	75	77	77	79	77	76	73	69	61	56	50	52	49		
4. Cincinnati, Ohio	1923	39	36	37	28	43	40	52	52	64	61	79	72	81	76	82	74	75	68	64	54	50	44	44	44		
5. Cleveland, Ohio	1923	36	30	33	24	36	35	40	46	47	54	59	71	66	71	69	69	68	65	65	52	52	42	43	40	68	72
6. Detroit, Mich.	1918	32	13	32	24	32	40	37	45	45	62	60	67	67	71	70	75	60	57	55	56	46	42	38	36		
	1919	33	31	32	29	35	36	41	46	51	57	...	74	71	75	70	70	66	67	59	56	44	39	33	23		
	1920	32	17	...	23	...	38	...	41	49	56	64	69	68	70	69	69	66	66	59	59	44	39	37	33		
	1921	32	30	32	30	36	43	44	53	49	62																
7. Flint, Mich.	1920	34	12	34	19	38	...	45	...	59	56	71	69	71	69	68	69	66	64	60	57	41	36	38	31	79	75
	1921	35	29	35	27	42	40	55	52	69	61	74	71	81	78	74	68	69	67	58	50	46	36	40	28	85	80
	1922	38	20	40	26	40	34	51	47	63	64	74	69	77	71	74	69	68	64	56	51	46	39	35	27	79	79
8. Grand Rapids, Mich.	1923	34	26	34	20	35	29	47	45	60	57	75	72	76	73	72	68	65	63	54	50	42	40	38	37	78	79
9. Minneapolis, Minn.	1923	34	18	34	10	35	21	42	44	58	60	72	71	76	76	69	67	64	63	54	48	40	39	35	30	81	77
10. New Orleans, La.	1915	43	52	46	56	48	55	57	69	73	77	79	84	82	85	82	83	77	81	71	73	65	66	51	57	84	84
	1916	46	61	45	57	49	64	55	68	66	77	75	82	79	82	83	83	81	79	71	72	61	63	50	57	82	84
	1917	47	60	46	59	54	66	61	68	67	72	77	80	82	83	85	83	81	78	72	66	61	59	54	52	82	87
	1918	48	48	51	63	58	69	63	68	70	76	82	83	83	83	85	82	82	77	73	75	66	62	61	58	84	86
	1919	47	51	46	57	52	64	62	68	67	74	76	80	87	83	89	83	82	80	83	80	67	66	50	57	91	91
	1920	49	56	50	56	51	60	60	69	67	78	77	81	83	82	86	81	84	82	74	71	65	58	54	54	86	88
	1921	53	59	52	60	60	71	64	68	70	74	82	81	88	83	88	84	87	83	74	71	67	67	55	61	90	91
	1922	48	56	46	63	49	62	63	73	71	76	79	82	83	82	84	83	83	81	75	72	66	67	55	64	86	85
	1923	46	61	46	57	50	62	59	70	68	74	77	80	82	80	84	82	81	81	72	70	57	60	52	61	84	86
11. Pittsburgh, Pa.	1912	...	20	34	25	36	35	48	53	61	64	70	68	75	73	70	69	68	69	57	56	45	44	36	36	81	75
	1913	37	40	34	29	39	43	46	52	61	61	70	70	75	74	77	73	68	65	59	54	46	45	39	36	81	81
	1914	34	34	36	24	37	37	46	49	59	63	75	71	77	74	75	73	68	64	61	58	45	43	37	30	79	77
	1915	34	31	37	37	39	33	48	56	61	58	70	67	72	72	72	69	72	69	57	56	45	45	36	32	77	77
	1916	36	38	36	27	36	34	46	49	59	63	64	65	79	77	79	74	66	64	55	55	43	44	36	33	82	82
	1917	34	32	34	27	37	41	48	50	55	55	66	68	73	74	77	72	66	62	55	49	39	40	34	24	82	84
	1918	34	19	34	33	41	45	50	49	66	68	72	68	75	72	79	77	64	60	55	58	45	44	39	41	82	84
	1919	36	34	34	34	43	42	50	51	55	61	79	75	79	75	75	70	70	66	63	61	46	43	36	28	82	79
	1920	34	24	34	28	37	43	46	47	59	60	70	68	75	70	75	72	70	67	64	60	46	42	37	36	77	77
	1921	32	35	34	36	45	51	57	57	64	62	77	73	82	78	79	70	75	71	57	54	48	45	41	34	86	81
	1922	34	29	34	35	39	43	50	53	64	65	75	71	79	74	75	71	72	70	59	56	46	45	36	36	81	79
	1923	33	33	33	27	37	39	47	50	55	60	76	72	77	73	76	71	72	67	60	52	46	43	42	43	81	81
12. Pittsburgh region:																											
a	1923	35	33	38	27	41	39	48	50	60	60	74	72	76	73	75	71	69	67	58	52	44	43	41	43		
b	1923	40	33	40	27	45	39	52	50	64	60	78	72	79	73	80	71	77	67	71	52	54	43	46	43		
c	1923	36	33	36	27	38	39	48	50	60	60	74	72	76	73	77	71	72	67	62	52	48	43	42	43		
13. Quincy, Ill.	1913	34	29	34	25	36	37	51	54	64	65	74	77	80	82	80	83	71	70	58	54	46	49	40	38	82	82

	1914	35	34	34	23	37	39	50	54	66	68	77	79	80	82	79	79	70	69	...	60	...	45	...	24	84	81
	1915	...	24	...	38	...	35	63	61	65	62	72	70	78	75	75	71	73	70	57	59	47	47	35	30	81	79
	1916	34	28	34	27	38	41	51	52	62	65	70	70	85	85	83	79	71	67	58	56	44	45	36	27	88	88
	1917	34	28	34	23	38	41	49	49	60	58	70	71	80	79	79	74	72	68	54	47	44	43	35	21	84	84
	1918	34	12	34	29	40	48	51	48	66	68	76	76	77	78	81	82	68	65	62	61	48	46	40	41	81	82
	1919	34	34	36	35	42	46	54	56	61	64	79	78	84	82	81	77	76	74	62	59	45	42	34	26	84	84
	1920	34	26	34	33	39	45	50	50	64	64	78	76	80	78	79	76	75	72	63	64	44	42	37	35	81	81
	1921	34	36	37	40	47	52	55	57	67	69	79	80	84	84	77	79	74	74	58	59	45	43	37	36	86	82
	1922	34	28	34	35	40	45	51	57	67	69	79	78	79	78	78	79	73	73	61	61	49	47	36	34	81	81
	1923	35	37	34	29	37	39	51	54	63	65	78	76	82	81	79	78	70	70	58	54	44	46	39	41	86	84
14. St. Louis, Mo.	1921	...	...	...	...	...	...	57	58	68	68	80	78	85	83	80	78	74	74	61	60	49	47	39	39	87	83
	1922	34	30	36	36	44	46	55	58	68	69	78	78	79	79	80	79	74	74	61	62	51	49	38	37	83	83
	1923	38	40	35	30	43	41	51	55	64	64	77	75	81	81	80	78	72	69	60	55	48	48	43	44	85	85
15. St. Joseph, Mo.	1923	34	37	34	29	34	37	48	54	64	62	72	74	78	79	77	76	68	69	61	52	43	45	36	37		
16. Springfield, Mass.	1920	33	19	33	24	33	35	37	45	48	56	59	66	64	70	64	72	59	58	54	...	42	...	36	...	66	66
	1921	33	27	33	30	36	42	44	53	52	58	58	...	66	74	65	...	65	...	54	53	43	40	36	...	69	68
	1922	33	...	33	29	34	38	39	...	51	...	61	...	66	74	65	72	61	68	52	57	42	43	35	37	70	67
	1923	33	23	33	23	33	34	39	48	50	57	60	68	65	69	65	68	63	63	52	50	40	39	37	40	66	66
17. Terre Haute, Ind.	1919	...	34	...	34	...	44	...	54	...	61	81	76	84	80	81	74	75	71	66	60	...	42	...	27	88	86
	1920	...	24	...	31	...	44	52	48	66	63	73	72	77	75	73	73	70	71	61	62	...	42	...	34	82	79
	1921	...	36	...	38	...	51	54	57	66	67	79	78	81	81	75	74	70	72	55	57	...	46	...	37	86	81
	1922	...	27	...	34	...	44	52	56	63	68	70	75	72	76	73	76	66	72	54	59	...	46	...	34	75	77
	1923	...	36	...	27	...	40	48	52	63	62	75	73	72	78	73	74	70	67	54	54	...	45	...	43	86	82
18. Toledo, Ohio	1917	33	26	33	22	40	38	50	45	54	53	69	66	75	73	76	71	67	62	49	46	42	39	34	22	82	84
	1918	34	14	34	28	40	42	49	46	66	65	74	68	74	72	78	76	62	58	56	56	46	43	39	38	79	80
	1919	33	33	32	31	40	37	49	47	58	57	77	74	77	76	73	71	70	68	60	57	44	40	34	24	80	77
	1920	34	18	34	25	39	40	45	42	59	57	71	70	74	71	73	70	70	67	60	61	44	40	38	33	77	77
	1921	34	32	34	32	46	45	54	54	64	62	77	73	82	79	74	71	73	70	56	53	43	40	37	32	88	79
	1922	34	24	34	31	38	38	51	49	65	63	74	70	76	72	75	71	71	68	58	55	46	43	36	30	79	79
	1923	34	30	34	23	36	34	47	47	58	55	76	72	76	73	73	70	68	65	56	51	44	41	39	39	79	79
19. Topeka, Kans.	1923	42	39	31	30	58	39	52	54	71	62	71	74	82	79	82	77	77	70	68	53	39	46	37	37		
20. Wilmington, N. C.	1923	49	50	48	46	58	57	63	61	72	68	81	77	84	78	84	79	80	74	68	62	57	53	51	54	86	86
21. Youngstown, Ohio	1918	42	15	36	30	46	41	59	48	75	64	83	67	91	70	93	75	84	57	79	55	61	42	48	38	97	102
	1919	45	32	54	32	52	39	...	48	68	58	91	73	96	74	90	69	86	61	72	59	63	41	50	25	99	95
	1920	50	20	54	25	50	...	61	44	70	57	81	67	86	69	90	70	90	65	82	58	68	40	52	32	93	95
	1921	50	32	45	32	54	47	63	54	75	61	90	71	93	77	90	69	88	69	75	53	55	42	48	32	99	97
	1922	46	25	45	33	50	40	61	50	75	62	90	69	91	72	93	69	91	66	88	55	77	43	64	32	100	102
	1923	50	30	46	25	48	36	63	47	77	56	93	68	99	70	99	69	97	64	88	...	68	41	50	41	102	104

1. Gunpowder River impounded for public water supply of Baltimore, Md.
2. Cahaba River impounded for public water supply of Birmingham, Ala.
3. Tennessee River at waterworks intake, Chattanooga, Tenn.
4. Ohio River at waterworks intake, Cincinnati, Ohio.
5. Lake Erie at waterworks intake, Cleveland, Ohio.
6. Detroit River at waterworks intake, Detroit, Mich.
7. Flint River at waterworks intake, Flint, Mich.
8. Grand River at waterworks intake, Grand Rapids, Mich.
9. Mississippi River at waterworks, Minneapolis, Minn.
10. Mississippi River at Carrollton filtration plant of municipal waterworks, New Orleans, La.
11. Allegheny River at Aspinwall filtration plant of municipal waterworks, Pittsburgh, Pa.
12. a, Allegheny River at Colfax, Pa.; b, Monongahela River at Rankin, Pa.; c, Ohio **River at Brunot Island. Water temperature furnished by the Philadelphia Co. of** Pittsburgh, Pa., from records at plants of the company. Air temperature at Pittsburgh, Pa.
13. Mississippi River at waterworks intake, Quincy, Ill.
14. Mississippi River at waterworks intake, St. Louis, Mo.
15. Missouri River at waterworks, St. Joseph, Mo.
16. Westfield Little River impounded for public water supply of **Springfield, Mass.**
17. Wabash River at waterworks, Terre Haute, Ind.
18. Maumee River at waterworks, Toledo, Ohio.
19. Kansas River at waterworks, Topeka, Kans.
20. Cape Fear River at waterworks, Wilmington, N. C.
21. Mahoning River at waterworks, Youngstown, Ohio. **Temperature of filtered water. Air temperature at Warren, Ohio.**

water from 450 to 1200 gal. per min. by substituting an air lift for the plunger pump. In the air lift compressed air is brought into the discharge pipe some distance below the surface of the water in the well, preferably supplied in finely divided bubbles. As the air rises the bubbles act as pistons, and before the flow begins the air serves to lighten the column in the discharge pipe while afterwards the air acts both as a means of lessening the fluid density of the mixture of the air and the water in the discharge pipe and of furnishing motive power required to keep up the continuous flow of water in the discharge pipe. However, the "slip" increases with the size of the air bubbles, and it seems best to supply the air to the discharge pipe through small holes. The overall air lift efficiency is as low as 20 per cent, and the installation has high initial and operating costs but it has the advantage of being able to pump readily even silty and gritty water. Figure 186[4] gives an idea of what can be done with the lift. The diameter of the air pipe which gives the best results has a ratio of 1 : 2 to 1 : $2\frac{1}{2}$ or 3, with the diameter of the discharge pipe, and the velocity of the water at discharge varies from 18 to 26 ft. per sec., whereas the velocity at the bottom is as low as from 6 to 8 ft. per sec. A large pipe means a low velocity and small friction loss, but it means also a much increased slip. The drop of water level when

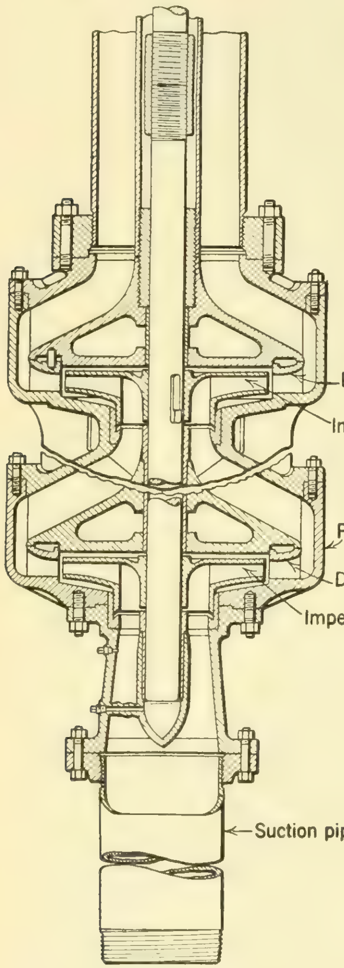


FIG. 187.—The Deep Well Type Centrifugal Pump.

[4] Blaisdell, Power, Nov. 23, 1920.

pumping begins varies, but usually is from 20 to 50 ft. The bottom of the well must be at least 10 to 25 ft. below the air pipe connection to the discharge pipe.

TABLE 77*a*

AIR-LIFT TESTS, POWER-HOUSE WELL

	Test Number			
	1	2	3	4
	Date of test			
	2/11/12			2/18/12
Length of 3-in. air pipe from discharge, feet	669.0	669.0	669.0	669.0
Surface to water static level, feet	155.0	155.0	155.0	155.0
Discharge to water static level, feet	177.0	177.0	177.0	177.0
Air pressure, starting, pounds	213.0	213.0	213.0	213.0
Air pressure, running, pounds	153.0	153.0	153.0	153.0
Friction (estimated) air pipe, pounds	11.9	10.0	8.6	10.2
Running pressure at nozzle, pounds	141.1	143.0	144.4	141.8
Head on air lift, feet	343.0	338.0	335.0	342.0
Submergence, running, feet	326.0	331.0	334.0	327.0
Submergence, running, ratio, per cent	49.0	50.0	50.0	49.0
Submergence, starting, feet	492.0	492.0	492.0	492.0
Submergence, starting ratio, per cent	73.5	73.5	73.5	73.5
Water velocity, feet per second	5.4	5.3	5.1	5.3
Water temperature, degrees F.	60.0	60.0	60.0	59.0
Drawdown, feet	166.0	161.0	158.0	165 0.
Water pumped, gallons per minute	1146.0	1132.0	1109.0	1129.0
Free air used, cubic feet per minute	2031.0	1847.0	1721.0	1880.0
Indicated horsepower, steam end compressor	383.0	380.0	326.0	360.0
Computed from above				
Indicated horsepower hour per 1000 gal. water	5.58	5.15	4.90	5.31
Specific capacity of well	6.9	7.0	7.0	6.8
Gallons per minute of water per indicated horsepower	3.0	2.9	3.4	3.1
Pounds of dry steam per 1000 gal. of water	84.9	78.3	80.9	80.7
Cubic feet of free air per pound of dry steam	21.0	20.8	20.9	20.5
Water horsepower	99.0	96.5	93.5	97.2
Least air horsepower (isothermal compression)	315.0	286.0	267.0	289.0
Efficiency of air pumping apparatus, per cent	31.4	33.7	35.0	33.6
Efficiency of compressor and pipe, per cent	82.4	81.7	82.0	80.4
Efficiency of air lift system, over all, per cent	25.8	27.6	28.7	27.0

The Centrifugal Pump.—The centrifugal pump (Fig. 187) is the latest development in deep-well pumping. These are specially designed with small outside diameters with thirteen or more stages. This design of pump has an efficiency of slightly less than three times that of the air lift, and as it has no valves, can handle even water containing silt and slime. The pump is designed for direct connection or belt drive. A summary of tests performed at Armour & Co. is as follows:

TABLE 77*b*

Date of Test	April, 1921	March, 1922	March, 1922
Static water level	234.0	280.0	280.0
Operating water level	308.0	338.0	314.5
Drawdown	54.0	58.0	34.5
Gallons per minute	1600.0	1470.0	1910.0
Total working head	330.0	365.0	336.0
I.hp. (input)	197.5	205.6	834.0
Gallons per minute per i.hp.	8.1	7.1	2.3
Efficiency, overall	67.5	65.9	24.7
Air pressure (operating)	....	....	156.6
Submergence running (per cent)	....	....	46.0

General Water Pumping.—For general pumping it has been found that, because of the almost universal application of electrical power and because of its own inherent advantages, the direct connected centrifugal pump has replaced the reciprocating pump to a large extent. The centrifugal pump is compact, small sized for its capacity and needs very little attention, and it has an efficiency above 70 per cent at its rated capacities. However, it is not a positive acting machine in the sense of a displacement pump. In its operation water is thrown out by the action of the revolving impeller as a result of which a particular velocity is imparted to the water, and the theoretical head becomes

$$h = \frac{v^2}{2g},$$

where

h = the head in feet;
v = velocity of the water in feet per second.

There are two forms of centrifugal pumps, the volute (Fig. 188) type, which permits the water leaving the impeller to come to partial rest in the spiral casing without the use of guide vanes, and the turbine type,

which has diffuser blades in order to slow up the velocity of the water without impact or shock. The volute type is the more common, and is the better design for variable conditions, while the diffuser blade design will give better economy at the speed and capacity for which it was designed.

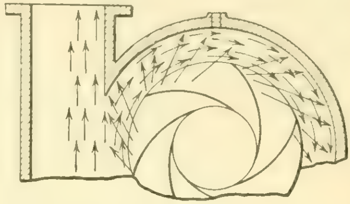


FIG. 188.—Volute Type Centrifugal Pump.

Characteristic curves are shown in Figs. 189 and 190, in which are given the variation of the head pressure with the output in gallons per minute, the efficiency and the required horse power. Figure 191 also

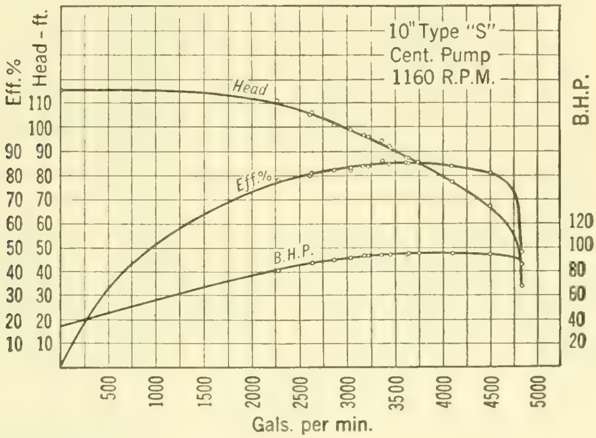


FIG. 189.—Centrifugal Pump Characteristic Curves.

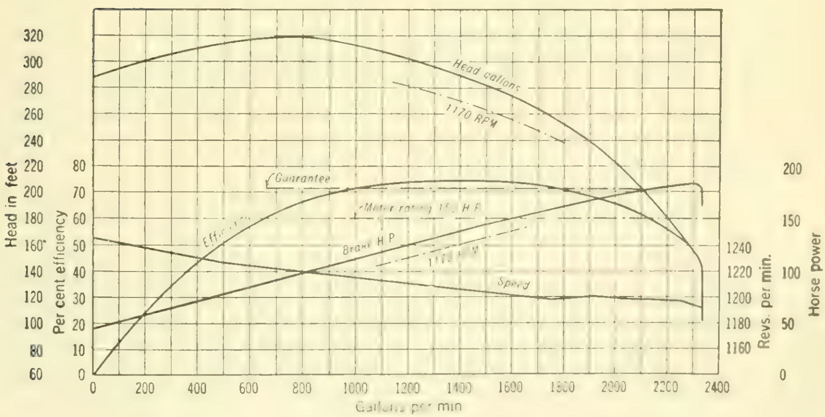


FIG. 190.—Characteristic Curves—Centrifugal Pump.

gives the effect of changing the speed, or the diameter of the impeller, showing that the pressure developed is proportional to some power of the peripheral velocity of the impeller. From experiment it follows that

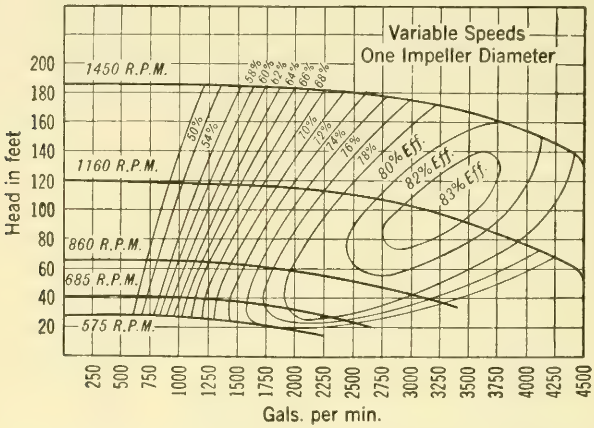


FIG. 191.—Change of Head with Variable Speed—Centrifugal Pumps.

if the diameter of the impeller is constant and the number of revolutions change:

1. The quantity flowing varies as the speed of the pump (the revolution per minute).
2. The head developed varies with the square of the speed.
3. The power required varies as the cube of the speed.

Likewise, if the speed is constant and the diameter of the impeller is changed:

1. The quantity flowing varies as the diameter.
2. The head varies as the square of the diameter.
3. The power required varies as the cube of the diameter.

At times the engineer can vary the speed and the diameter of the impeller so as to get the head and the quantity of liquid pumped to suit best the conditions in the plant.

The Cooling Tower.—Where the cost of water is excessive, or where water must be conserved for any reason, advantage may be taken of the cooling tower or the spray method of cooling. The cooling tower as a rule can give a temperature of the water to within three to five degrees of the wet bulb temperature. The spray nozzle can not do so well with a single set of sprays, as usually the maximum range of cooling is from five to eight degrees. In both methods of cooling the principle is to bring the water into intimate contact with air

in order that by the action of evaporation of some of the water into the air the remainder may be cooled.

The cooling tower generally makes use either of a wetted surface with the water flowing by gravity over these surfaces and the air ascending by natural draft, or, simply, the lateral movement of the air through the tower, or of a tower arrangement for securing fine drops of water

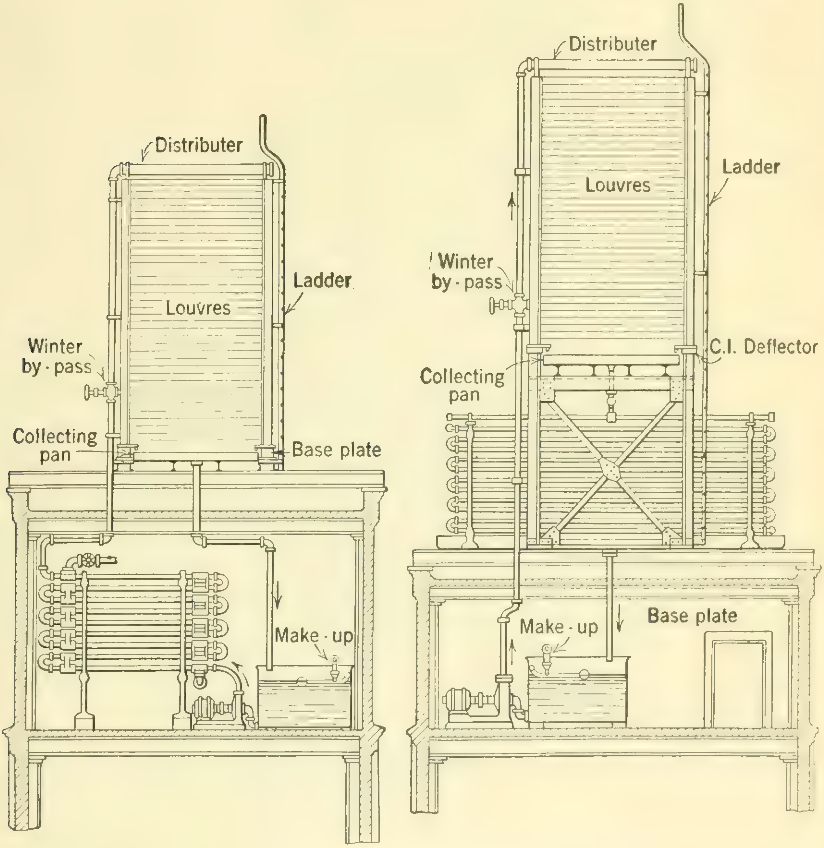


Fig. 192.—Cooling Towers with Ammonia Condensers.

and free fall. This means in refrigeration the placing of the tower in an exposed position in order to get the best possible air flow (Fig. 192). The spray nozzle (Fig. 195) depends on the atomizing of the water by means of a suitable nozzle, using from 5 to 15 lb. per sq. in. water pressure. In either case the cooling of the water is through the ability of the air to evaporate water, the limit of evaporation being the wet

bulb temperature. Some heat can at times be absorbed by the air, but in refrigeration this amount is relatively small because condenser cooling water is seldom much in excess of 100 deg. F. Assuming the latent heat of evaporation to be approximately 1000 B.t.u., the amount of make-up

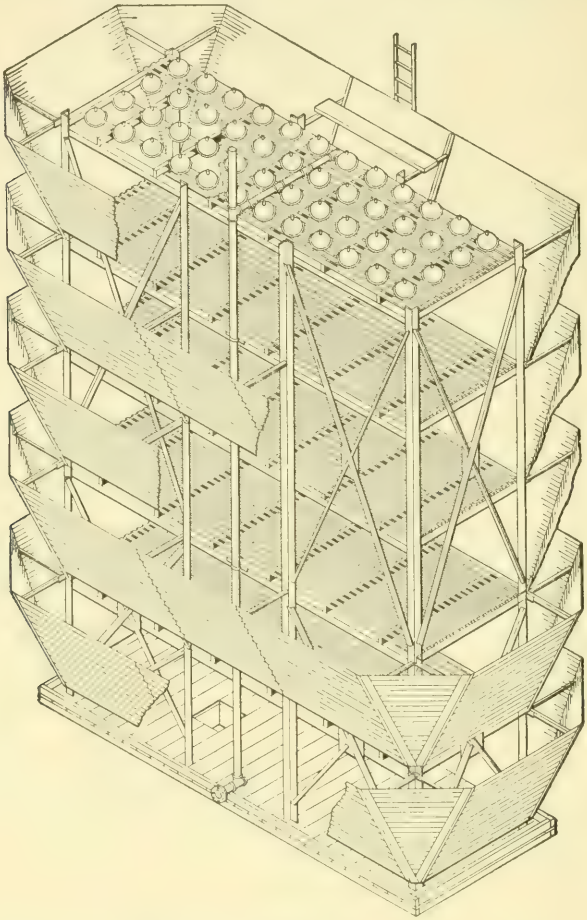


FIG. 193.—Cooling Tower.

water required will be 1.0 per cent for every ten degrees cooling of the water, not allowing for the cooling due to the air.

Construction.—The cooling tower is usually constructed with steel frames. The cooling surfaces or decks are made of special anti-corroding metal, structural steel, or of lumber, such as cypress or redwood. The tower is designed with louvres all around, usually designed to catch the

spray caused by the wind and return it to the tower, for unless care is taken in this respect the loss due to windage becomes excessive. The tower is placed sufficiently high as to permit the water to flow by gravity over the condenser if the condenser is of the atmospheric type. The pumping head is that necessary to lift the water up to the dis-

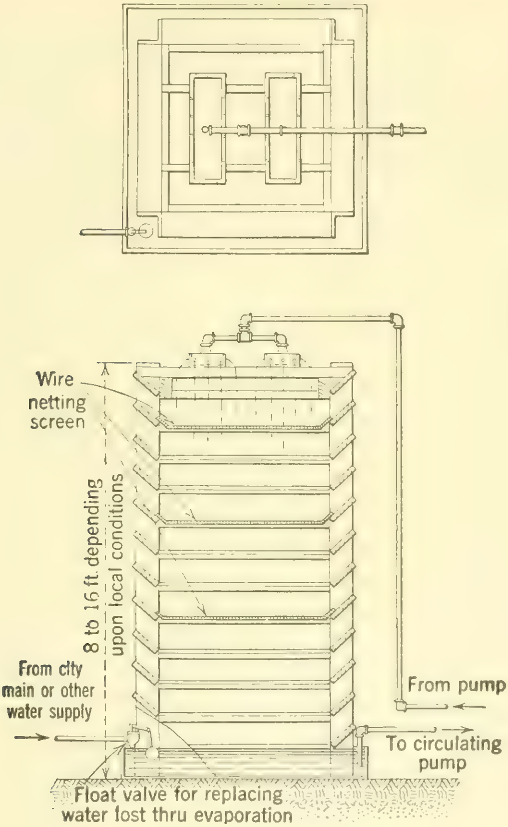


Fig. 194.—Cooling Tower.

tributing troughs. Types of construction are shown in Figs. 192 to 194. If desired, the circulating pump can be placed in the engine room, and if care is taken to make use of the principle of the syphon the only head required on the pump will be that due to the pipe friction, velocity head and the lift from the well to the top of the tower. Table 79 shows the approximate size of the usual tower, and Table 78 gives results of tests on cooling towers.

In the spray nozzle system the pumping head is that necessary to lift

TABLE 78

TYPICAL TEST ON A "BURHORN" COOLING TOWER

Date	Time	Water Temperature, Degrees		Air Temperature, Degrees		Direction of Wind
		To tower	From tower	Wet bulb	Dry bulb	
July 10	4:00 p.m.	87	70	68	84	N. W.
July 11	3:30 p.m.	78	68	65	88	W.
	4:00 p.m.	78	65	64	85	W.
	4:15 p.m.	74	66	64	84	W.
July 14	11:30 a.m.	73	65	62	78	N. W. Heavy wind
	1:45 p.m.	76	69	67	81	N. W. Wind slight
	4:45 p.m.	77	70	67	82	S. W. Slight
July 15	7:30 a.m.	75	68	64	71	W.
July 17	3:15 p.m.	79	73	70	81	S. E.
	4:15 p.m.	79	73	70	86	S. E.

TABLE 79

Capacity, Gallons per Minute	Dimensions, Inches		Piping, Inches		Weight Complete, Pounds
	Footings	Overall	Supply	Discharge	
8	. . . .	$3\frac{1}{2}\times 7\times 10$	$\frac{3}{4}$	$\frac{3}{4}$	1,000
25	4×4	$7\times 7\times 15$	$1\frac{1}{2}$	2	2,800
50	4×8	$7\times 11\times 15$	2	$2\frac{1}{2}$	4,600
75	6×9	$11\times 14\times 20$	$2\frac{1}{2}$	3	13,400
100	6×12	$11\times 17\times 20$	3	$3\frac{1}{2}$	15,000
125	6×15	$11\times 20\times 20$	$3\frac{1}{2}$	4	18,000
150	6×18	$11\times 23\times 20$	$3\frac{1}{2}$	4	19,500
175	6×21	$11\times 26\times 20$	4	$4\frac{1}{2}$	21,300
200	13×12	$19\times 19\times 30$	4	5	32,600
300	13×18	$19\times 25\times 30$	$4\frac{1}{2}$	6	42,000
400	13×24	$19\times 31\times 30$	5	6	49,900
500	13×30	$19\times 37\times 30$	6	7	59,700
600	13×35	$19\times 42\times 30$	7	8	67,700
700	13×41	$19\times 48\times 30$	7	8	77,500
800	13×47	$19\times 54\times 30$	8	10	85,200
900	13×53	$19\times 60\times 30$	8	10	95,900
1000	13×59	$19\times 66\times 30$	10	12	112,700

the water up to the nozzles plus the head necessary to form the spray. The pressure required at the nozzle to form the spray is kept as low as possible, and it should not exceed 10 lb. per sq. in. As the water in spray form will drift excessively in the wind, louvres are essential. Even with the use of louvres the loss becomes very great at times.

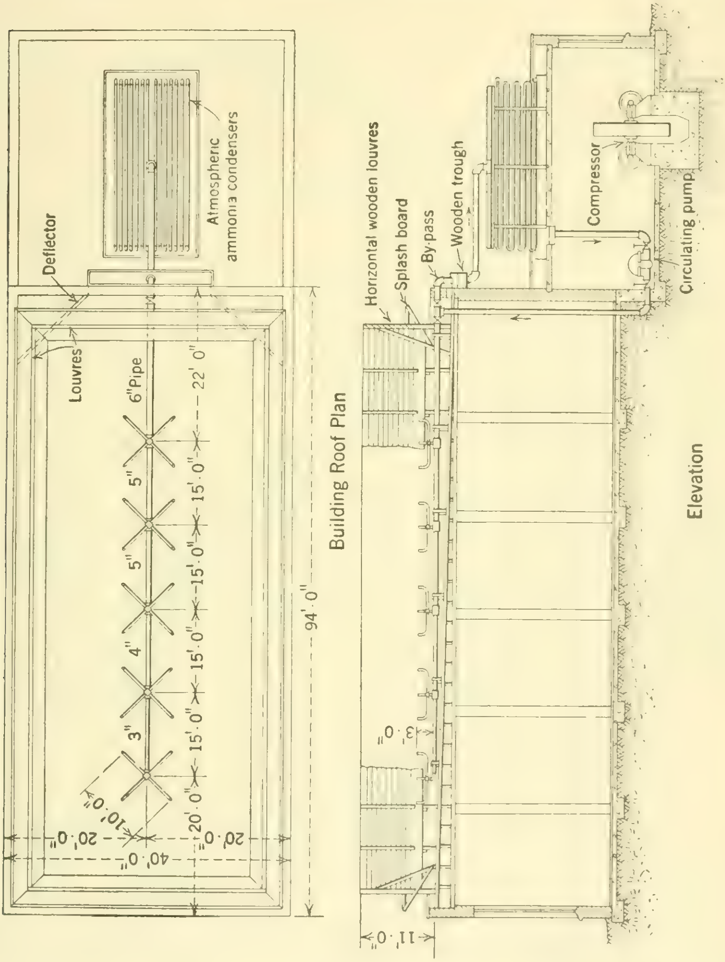


FIG. 195.—Sprays on Roof with Atmospheric Ammonia Condensers.

Figure 195 gives a typical arrangement of sprays on the roof, from which the water flows to the condenser.

Water Cooling Formula.—The following equation expresses the sensible heat gained or lost from the air in contact with water:

$$Q_1 = M_a c_p (t_2 - t_1),$$

where

M_a = weight of air, in pounds;

c_p = the specific heat of air;

t_2 and t_1 = the final and the initial temperatures of the air.

The amount of heat gained or lost from the vapor in contact with the water:

$$Q_2 = \left(\frac{M_1 + M_2}{2}\right)c_p(t_2 - t_1),$$

where

M_1 = initial amount of water vapor in the air;

M_2 = final amount of vapor.

The amount of heat absorbed during partial evaporation:

$$Q_3 = (V_2 - V_1)r,$$

where

V_1 and V_2 = the vapor content of the air entering and leaving the system;

r = the latent heat of vaporization.

The amount of the cooling of the water is:[5]

$$Q = M_w(t_2 - t_1),$$

where M_w = the weight of the water.

Coffey and Horne[6] found an expression for the heat flow to and from the wet bulb, which gives a measure of the cooling effect to be found in cooling towers and spray ponds. This expression is:

$$E = (e_T - e_t) + 0.0116(T - t),$$

where

E = measure of the effective cooling head, in inches of mercury;

T = temperature of the water being cooled;

t = temperature of the wet bulb, deg. F.;

e_T = vapor pressure, at temperature T, inches of Hg;

e_t = vapor pressure, at temperature t, inches of mercury.

Figure 196 shows these relations graphically.

The Optimum Water Rate.—In general the refrigerating engineer has not used any scientific method in arriving at the amount of water to be used in the plant. It is generally understood that an increase in the amount of the condensing water means a reduced condenser pressure, and therefore a smaller power consumption by the compressor. How-

[5] Water Cooling System Efficiency, V. J. Azbe, Mechanical Engineering, 1924.

[6] Coffey and Horne, A. S. R. E. Journal, Vol. I, No. 1.

ever condenser water has usually either to be pumped from a deep well, into a cooling tower or through sprays. As the amount of water used by the condensers increases the pumping cost also increases. Doubling the amount of water so as to operate with a 6 deg. F. instead of a 12 deg. F. range of temperature may reduce the condenser pressure from 158.0 to 139.2 lb. per sq. in. Whether this increase in the amount of water is justifiable will certainly depend on the total cost of pumping the water. In all likelihood the engineer will be inclined to set on a certain pressure as the operating maximum, say 185 to 190 lb., and when

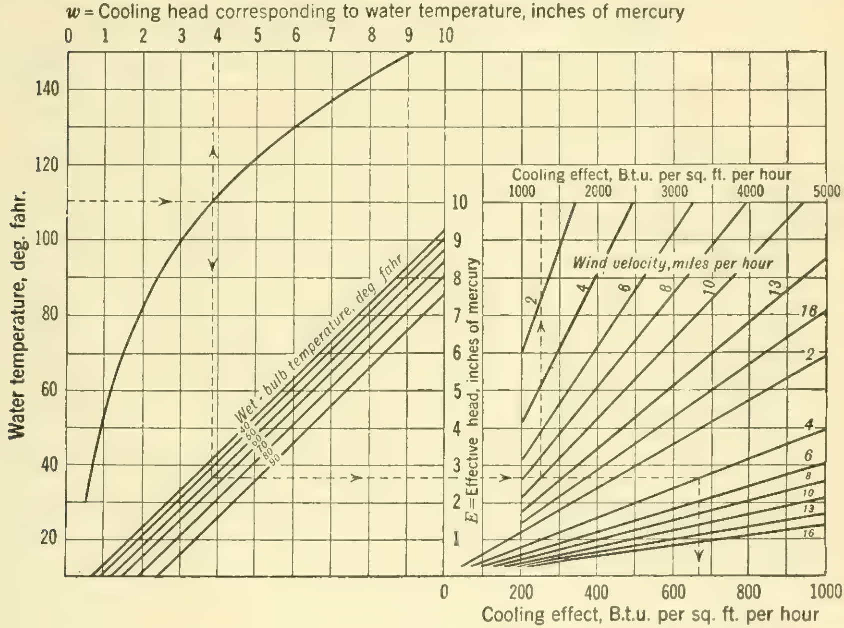


FIG. 196.—Graphical Method of Securing the Cooling Effect.

the condenser pressure becomes greater than this he will attempt to pump more water.

In the following some attempt is made to show how an economical water rate may be determined if certain operating costs are known:[6a]

Purchased Water.—The total cost of purchased water is

$$C = \frac{Amt_d}{E_t} + \frac{60BH_c}{1000 \times 8.33t_d}$$

[6a] T. K. Sherwood, Economic Balances in the Design and Operation of the Ammonia Compressor, Refrigerating Engineering, Feb., 1927. R. W. Waterfill, Balancing Compressor Power and Condenser Water Costs, Refrigerating Engineering, Sept., 1927.

where A = the cost of power per hp.-hr.

m = hp. increase per ton of refrigeration per degree increase of condenser temperature

t_d = temperature difference of the condenser water in and out of the condenser

E_t = Combined efficiency of the compressor and motor

B = Cost per 1000 gal. of condensing water

H_c = Heat to be removed from the condenser per ton per minute $= 200 + 42.4W_c$

W_c = The condenser i.hp. per ton of refrigeration

$G = H_c \div 8.33_d$ = the condensing water per ton.

In order to find the minimum cost the first derivative is equated to zero, and therefore

$$\frac{dC}{dt} = \frac{Am}{E_t} - \frac{0.06BH_c}{8.33t_d{}^2} = 0$$

from which the values for t_d and G may be obtained.

Pumped Water.—The total cost of pumped water is

$$C = \frac{mt_d}{E_t} + \frac{8.33G(F_s + f)}{33{,}000E_p}$$

Where E_p is the combined pump and motor efficiency

F_s is the static head on the pump.

f is the friction head due to the flow of the water in the pipes.

In order to make use of the equation f must be replaced by an expression in terms of the variable. Sherwood uses the expression, which is adopted by Waterfill,

$$f = K\left(\frac{D}{t_d}\right)^2$$

where K is a constant giving the value of f for an assumed temperature difference D

therefore

$$C = \frac{mt_d}{E_t} + \frac{H_c\left(F_s + K\left(\frac{D}{t_d}\right)^2\right)}{t_dE_p \times 33{,}000}$$

$$\frac{dC}{dt_d} = \frac{m}{Et} - \frac{H_cF_s}{t^2{}_dE_p \times 33{,}000} - \frac{3H_cKD^2}{t^4{}_dE_p \times 33{,}000} = 0$$

Comparison of Cooling Towers and Sprays.—The first cost of the spray system should be less than that of the cooling tower, but the loss of water due to drift will be greater with sprays. As a rule the tower

will have the advantage in congested regions where the local conditions are such that an elevated position, free from the wind protection of tall buildings, can be secured. The water in either case must be divided into small particles and the fall, especially in the case of the tower, must be interrupted frequently in order to keep the water in the air for as long a time as possible, whereas the air must be permitted free passage through the tower or over the spray pond.

TABLE 80

List of Nozzle Sizes

Nozzle Number	Pipe Size for Which Nozzle Is Threaded, Inches	Pressure in Pounds per Square Inch											
		5	6	7	8	9	10	12	15	20	25	30	40
		Capacity in U. S. gallons per minute											
60	2	53.0	58.0	62.5	67.0	71.0	74.5	82.0	92.0	106.0	118.0	130.0	150.0
50	2	44.0	48.0	52.0	55.5	59.0	62.0	68.0	76.0	88.0	98.0	107.0	124.0
45	2	40.0	43.5	47.0	50.0	53.0	56.0	61.0	69.0	79.0	88.5	97.0	112.0
40	2	35.4	38.8	42.0	45.0	47.5	50.0	55.0	61.0	71.0	79.0	87.0	100.0
35	2	30.8	33.8	36.4	39.0	41.2	43.5	48.0	53.5	61.5	68.0	75.0	87.0
30	$1\frac{1}{2}$	26.2	28.6	31.0	33.0	35.0	37.0	40.5	45.5	52.0	58.5	64.0	74.0
25	$1\frac{1}{2}$	22.0	24.0	26.0	27.8	29.4	31.0	34.0	38.0	43.8	49.0	53.5	62.0
20	$1\frac{1}{2}$	17.5	19.4	20.8	22.2	23.5	24.8	26.0	30.5	35.0	39.0	43.0	49.5
15	$1\frac{1}{2}$	13.0	14.3	15.5	16.5	17.5	18.5	20.2	22.6	26.0	29.0	32.0	37.0
10	$1\frac{1}{2}$	8.7	9.6	10.4	11.1	11.7	12.4	13.5	15.2	17.5	19.5	21.5	24.8
8	1	7.0	7.7	8.3	8.9	9.4	9.9	10.8	12.1	14.0	15.6	17.0	19.7
5	1	4.4	4.8	5.2	5.5	5.9	6.2	6.8	7.6	8.8	9.8	10.8	12.4
4	$\frac{3}{4}$	3.5	3.8	4.1	4.4	4.7	4.9	5.4	6.1	7.0	7.8	8.6	9.9
3	$\frac{1}{2}$	2.6	2.9	3.1	3.3	3.6	3.7	4.1	4.5	5.2	5.9	6.4	7.4
2	$\frac{1}{2}$	1.7	1.9	2.1	2.2	2.3	2.5	2.7	3.0	3.5	3.9	4.3	4.9
A	$\frac{3}{8}$	1.3	1.45	1.55	1.65	1.78	1.86	2.0	2.3	2.6	2.9	3.2	3.7
B	$\frac{3}{8}$	.95	1.04	1.12	1.20	1.27	1.35	1.48	1.65	1.9	2.1	2.3	2.7

CHAPTER X

ERECTION AND OPERATION

The refrigerating plant is subject to operating difficulties more than is the case with other comparable systems, because of inherent features in the cycle. In consequence of these difficulties, resulting in improper charging and operation, in an easy loss of capacity from a number of causes, in excessive condenser pressure, etc., it is desirable to consider all of these subjects in detail.

Foundations.—Foundations for refrigerating machines are required for two reasons—to make the compressor secure, and to absorb the vibrations developed. As considerable care is now being taken in the matter of balancing, vibration is no longer so important a factor as formerly. In this respect the twin vertical single-acting compressor is better than the horizontal double-acting or the duplex compressor. The necessary weight of the foundation being such a variable, it has to be worked out in each case by the individual designers. These foundation weights are made up for fair ground conditions. If the "soil" is rock or compact hard pan it is evident that the size and the weight of the foundation can be decreased, while if it is soft, wet clay or marshy soil, the footings under the foundation must be widened.

Template and Anchor Bolts.—A good mixture of concrete should be used, either a 1 : $2\frac{1}{2}$: 5 or a 1 : 2 : 4 mixture by volume of cement, clean sand and crushed stone. Sufficient time should be allowed for hardening before any weight is placed on the foundation, usually 36 to 48 hours or longer being required. Before pouring, a templet of the anchor bolts with the bolts and anchor plates attached should be set in place and rigidly fastened so that they will not float out of position. Care must be taken to insure the proper level, and the proper alignment with the building wall. The larger machines usually have pockets in the foundation so that access may be had to the anchor plates at any time. In any case ferrules, made of tin, old pipe or thin lumber are so placed that the anchor bolt will hang central and will have a clearance of $\frac{3}{4}$ to 1 in. all around it. While the concrete is green these ferrules can be removed. The anchor plates are screwed up to the middle of the threads on the lower part of the bolt. The pouring of the concrete should

stop at a point from $\frac{1}{2}$ to 1 in. below the base level in order to permit grouting under the machine.

Leveling and Grouting.—When the foundation is sufficiently hard to take the weight, the compressor may be moved into place and leveled. Wedges made of wood or iron should be placed near the anchor bolts, and the machine should be leveled in both directions. In a horizontal machine the position of the center of the shaft is required accurately, and this can be located at right angles to the compressor line of centers by means of a triangular templet made with distances between the apexes in the ratio of 3 : 4 : 5. The anchor bolt holes should be filled first with a thin grout of sand and cement in the ratio of 1 to 1 by volume. Then fill in underneath the machine with grouting thin enough to float but not to show water on top, making sure that the grout is worked in under all parts of the compressor bed. In a few hours the wedges can be removed and the foundation finished with a smooth finished surface. The foundation bolts must not be tightened until the grout is thoroughly dried (after two to three days, as a rule). Small compressors can be erected sometimes by drilling out the concrete and using an expansion bolt. These holes are filled with grouting as before.

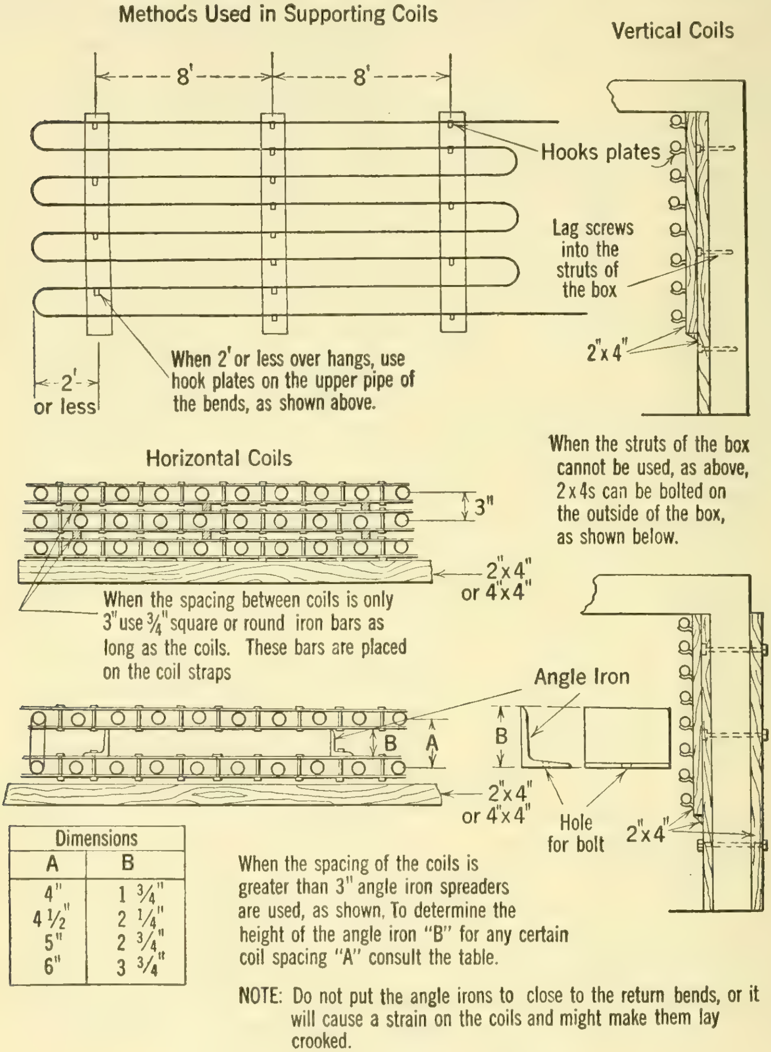
Piping.—Piping[1] is one of the important elements in refrigeration machinery. Formerly wrought iron was used extensively, and then, later, extra heavy pipe was specified. There is no standard practice at the present, but the use of full weight black steel pipe is quite common, even for condensers and the line from the compressor to the condenser. Yet some concerns continue to specify the use of extra heavy pipe for the evaporating coils in the tank. In general, the feeling prevails that steel pipe can be relied on much better now than, (say), 10 years ago, and that care in the erection and prevention of excessive corrosion have changed conditions so that extra heavy pipe is no longer justified.

The essential of good pipe work is clean, accurately cut pipe threads, using sharp dies with Briggs standard threads for work in the United States. These threads should be cleaned thoroughly before making up by brushing them with gasoline and wiping clean the pipe threads and the flanges or screwed fittings. Making up the joint so that it will be *hot* is no indication that it will be tight, as the heat indicates friction only, which, of course, could be developed by dirt or scale as well as by tightness.

Kind of Joint.—Both soldered and lithage joints are used, although the tendency appears to be towards the elimination of the former except in condenser practice where the soldering is done at the factory. The lithage joint is as easy to make up as a first-class steam fitting connection.

[1] See Chapter IV.

The solder joint is made by cleaning the fittings and the pipe until the metal is bright. The pipe and the fittings are then dropped into molten solder, made of half-and-half of tin and lead, and the threads



Dimensions	
A	B
4"	1 3/4"
4 1/2"	2 1/4"
5"	2 3/4"
6"	3 3/4"

Fig. 197.—Details of Pipe Coil Supports.

are evenly tinned over with the use of a soldering acid made of muriatic acid cut with zinc. The fitting is made up while both are hot, and the recess at the back of the fitting filled with solder. In the larger pipe sizes the solder in this recess should be caulked.

The lithage joint is made up by applying a thin layer of the mixture to the threads of both the pipe and the fittings. The mixture should be made in small amounts as needed in the proportion of $1\frac{1}{4}$ parts of lithage to 1 part of glycerine by volume, and it should be mixed thoroughly. Care should be taken not to have excessive lithage lest the effective area of the pipe be decreased, especially in the case of small pipe.

Much future trouble may be eliminated by care in initial cleaning of the inside of any pipe to be erected so as to get rid of any loose mill scale or loose pipe threads, and it is advised that at the very least the pipe be examined on the inside, and that it be rapped with a light hammer when held vertically. Bent pipe should be tested for obstructions by rolling a steel ball through the pipe.

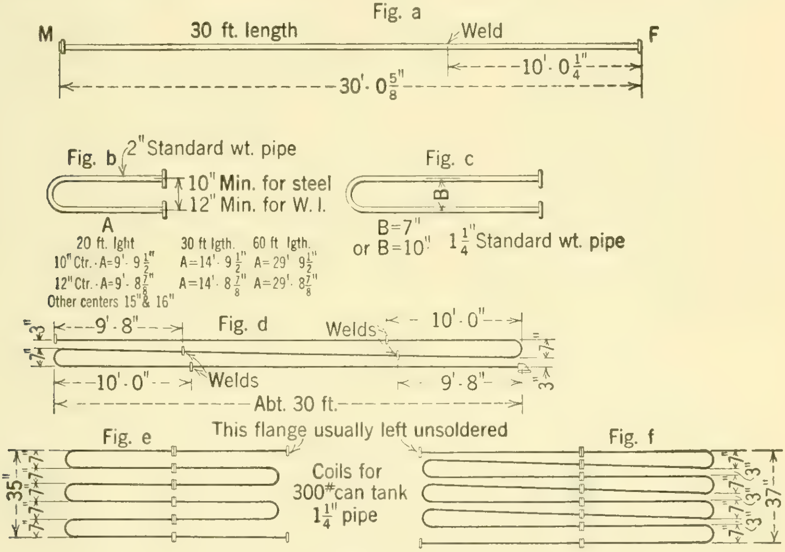


FIG. 198.—Details of Ammonia Coils.

Piping Details.—Figures 234 to 239 in Chapter XII give some details of piping, particularly for large installations. In addition, Figs. 197 to 201 give details more applicable to small plants. Piping practice is not standardized, and is given here only as a suggestion. The principal considerations are proper support, and slope of the pipe lines so that drainage of the water pipes and prevention of liquid seal in the ammonia pipes will be provided for, and the proper clearance allowed for so that erection or repairs will be as simple as possible.

The pipe should be very carefully erected, remembering that cold storage piping has a heavy accumulation of frost in most cases, and

that it may also be filled with liquid ammonia. Low points, where liquid may trap in the piping, should be eliminated as far as possible, unless

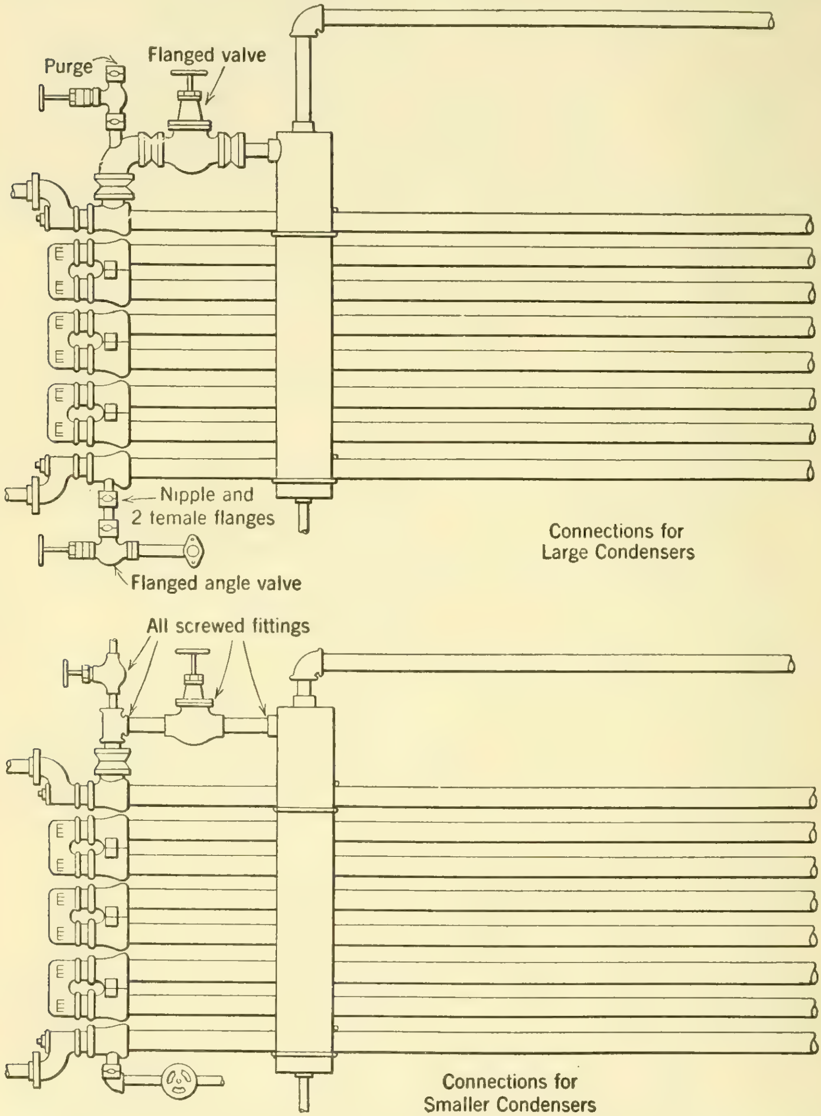


FIG. 199.—Ammonia Condenser Connections.

connection is made at every low point to suitable traps and to the regenerator. Pipe vibration must be kept at a minimum or leaks may

occur at any time. Flanged joints are made up with either lead or oil proof rubber gaskets, and it is wise to chalk-mark the flanges as the

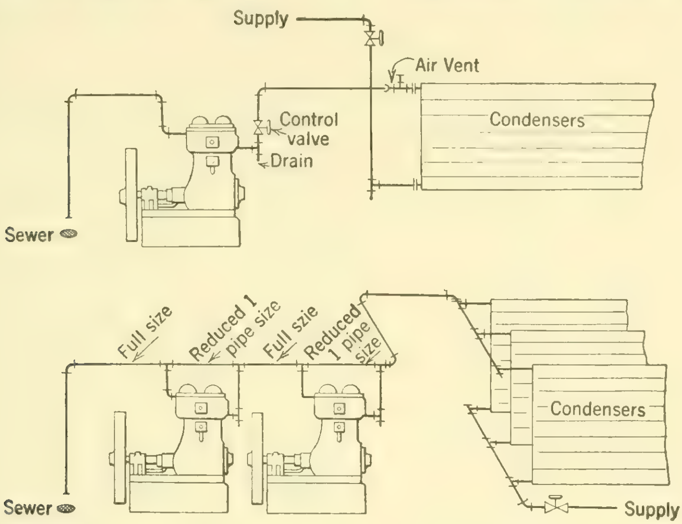


FIG. 200.—Cooling Water Connections.

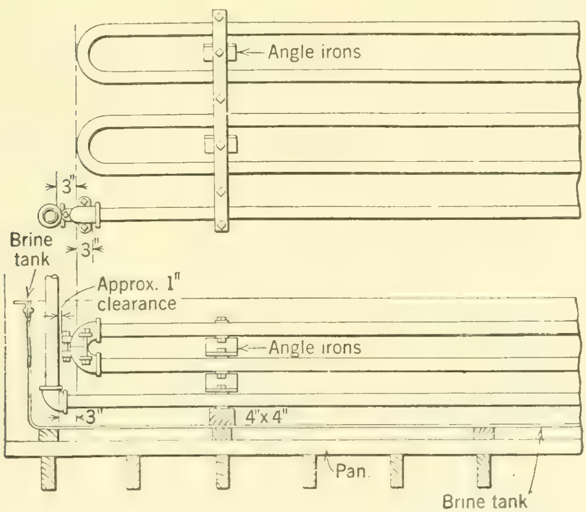
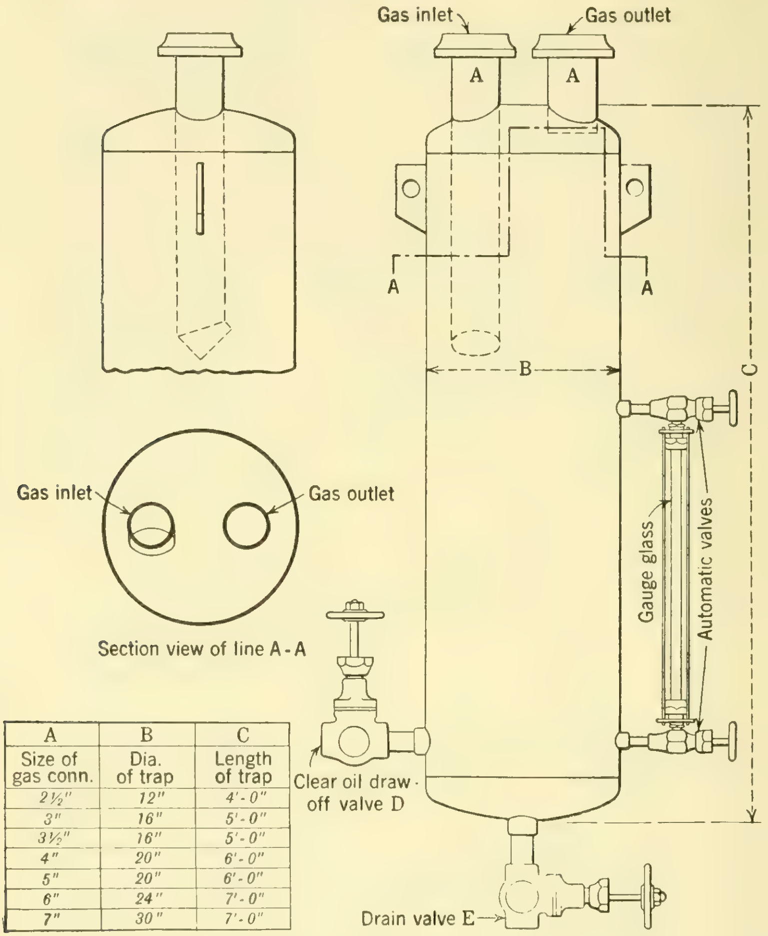


FIG. 201.—Ammonia Coils in Brine Tank.

gaskets are put in place, as a check. The difficulty with lead gaskets is that lead has no elasticity and that it will not give good service

when exposed to the hot discharge gases from the compressor. For the ammonia discharge lines it is better to make use of a high-grade asbestos sheet gasket material which takes without trouble the highest temperatures found in refrigeration and has some elasticity. In making



A	B	C
Size of gas conn.	Dia. of trap	Length of trap
2½"	12"	4'-0"
3"	16"	5'-0"
3½"	16"	5'-0"
4"	20"	6'-0"
5"	20"	6'-0"
6"	24"	7'-0"
7"	30"	7'-0"

Fig. 202.—Oil Separator.

up, the bolts must be pulled up evenly all round, and special care must be taken to see that the flange surfaces are free from dirt and scale.

The *oil separator* (Fig. 202) can separate the oil only when it is in a liquid condition. In consequence the separator should be placed as near the condenser as practicable, in order that the gas passing through

may become as cool as conditions will permit. At times it may be wise to install a precooler just ahead of the separator. The precooler is sometimes used in ice-making plants for the purpose of heating water for the drip tank.

The *scale trap* (Fig. 203) is placed in the line especially to protect the compressor. Even with the best of care in erection, mill scale, sand and metal cuttings from the dies will circulate with the ammonia or other refrigerant. The scale trap should be as close to the compressor as possible, and it should be of the size and the design required so that it will function well. At first the need of such a device is very great, but this need decreases as the time elapses. Means should be provided for easy cleaning of the trap with as little air as possible entering the system each time the trap is cleaned.

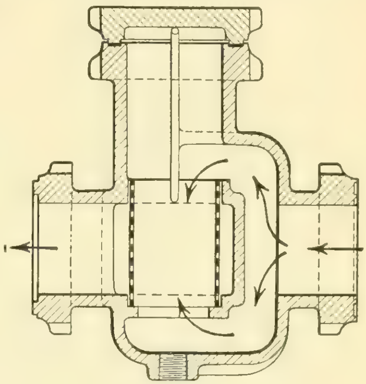


Fig. 203.—Scale Trap.

As soon as the piping has been erected, water and steam or electric connections made, the plant is ready to be tested out for leaks, using air under pressure. The compressor is used for pumping this air pressure, care being taken not to pump up too rapidly at any time for fear of explosions.

Make sure that the compressor is in good shape as regards rod packing, valve action, tightness of bearings, etc. See that the lubrication is in good shape and that the proper kind of oil is used in the cylinders.

Cylinder Oil.—It is very important that the proper ice machine oil is used for the cylinder and the crankcase in the enclosed type of compressor. As a rule a freeze test of -20 deg. F. is sufficient, but in some cases—the carbon dioxide compressor when working on low temperatures, for example—it is worth while to get a lower freezing test temperature for the oil, say, -30 deg. F. The splash type of enclosed machine should be filled with oil up to the mark on the frame of the compressor.

Starting Up.—All refrigerating machines are designed so that the compressor may be used to pump out the system normally under condenser pressure and to discharge this gas into the low pressure side. This is accomplished by means of by-pass valves cross connected (Figs. 204 and 205). Most compressors are also fitted with a by-pass from the discharge into the suction, especially motor driven compressors, in order

that the required starting torque shall be as small as possible. Each cylinder also has a stop valve for the suction and the discharge line close up to the cylinder. There is required also an opening (a valve or a plug) to the atmosphere on the suction and the discharge side of the cylinder. All of these connections are required at different times during the operation of the compressor.

Pumping an Air Pressure.—Open the connection to the atmosphere on the suction side and close the suction stop valve. To put pressure on the high side only, keep the expansion valve closed; to test out the entire system, have the expansion valves open, and to test out the low

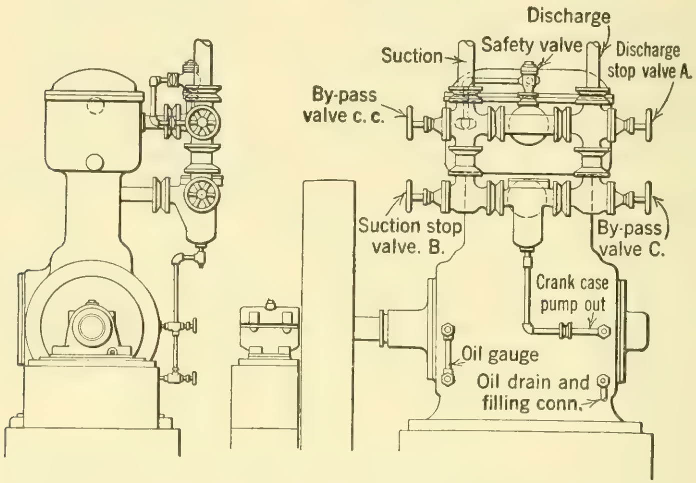


FIG. 204.—By-Pass Valves for the Vertical Single Acting Compressor.

pressure side only, have the cross connections open—with the discharge open to the atmosphere—and the expansion valves closed.

With water passing through the jackets operate the compressor slowly (or intermittently) and raise the pressure on the system to about 150 lb., unless large leaks evidenced by the noise of escaping air appear before this pressure is reached. In a new plant the first things to look for are split pipe, flanges with defective or even without any gaskets, leaky valve bonnets, valve stems, etc. At first the escape of air can be heard, but when that is no longer true painting with soap and water must serve. The latter is only good for small leaks—all others can be heard if there are no other noises in the compressor room. The lather should be put on the pipe and fittings freely, and the pressure raised to 200 lb. at this point, for the entire system, safeguarding the low pressure gage by shutting it off.

When all leaks have been located and repaired the next step is to make a duration test by pumping up to 200 lb. and allowing the pressure to remain on for 12 to 24 hours. Any drop of pressure not accounted for by a lowering of the temperature of the air is due to leaks and should not be allowed to be appreciable.

Blowing Out the Coils.—The piping is usually designed so as to permit the *blowing down* of each set of coils to permit freeing the pipes of any loose dirt, steel chips, etc., which may be in the line. For this purpose use pressures of 250 to 275 lb. and open freely to the atmosphere so that a high air velocity will be acquired. Continue the process until the whole system has been blown down. After blowing down it is well to go over the compressor thoroughly in order to clean out any grit or dirt which has worked into the cylinder (the crankcase of the enclosed machine) and the valves and ports. It is valuable to repeat this process about a week after charging and starting operation.

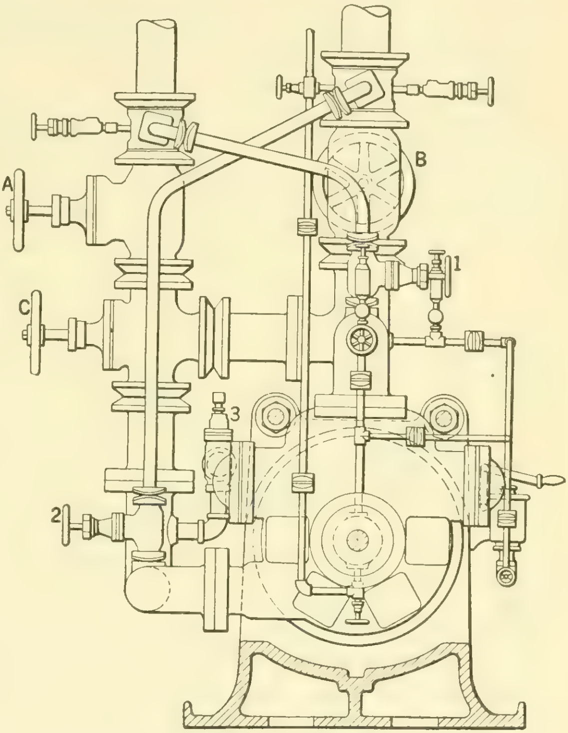


Fig. 205.—By-Pass Valves for Horizontal Double Acting Compressor.

Test with Ammonia.—Pump a vacuum on the entire system. This can be done usually by closing the cylinder discharge stop valves (and the by-pass valves), opening all lines and expansion valves and opening the discharge to the atmosphere by means of the valve or plug arranged for that purpose. Pump as good a vacuum as possible, as it is easier, and cheaper, to free the pipe system of air at this point than after charging.

Attach the ammonia charging drum to the charging connection and permit liquid to flow slowly into both the low pressure and the high pressure side. Permit the pressure to rise to about 50 lb. and then look for leaks, first by the smell and the noise of the escape, and then by means of the sulphur stick. When the sulphur dioxide gas formed by the burning of the sulphur stick comes in contact with ammonia gas there is formed a *white* smoke, the indication of an ammonia leak. These sticks may be in the form of candles with a small wick in the center, or they may be sulphur-coated heavy twine—the latter prepared by dipping the twine in liquid sulphur. Even pieces of soft wood may be dipped into the liquid sulphur to form a convenient "candle." Small leaks around welded joints can be stopped by caulking and with care this can be done also to malleable fittings. Bad leaks may require the pumping down and the replacing of the defective part. If the bad leaks are only in one part of the system it may be easier to pump down that

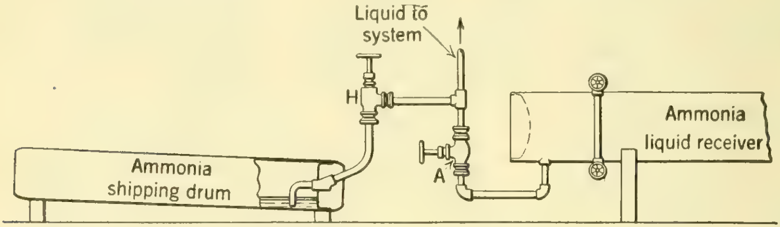


Fig. 206.—Charging Connections.

part only and, storing the ammonia in the other part of the system, to break into the line that needs repairing. The larger plants have separate pump-out compressors and pump-out lines to the different coils or sets of coils, so that repairs can be made without interfering with the operation of the other parts of the plant. The smaller installations, however, need to use the compressor, and the cross connection by-pass valves, to pump out from one part and to store the ammonia in the other part of the system during which operation the plant must be out of commission.

To Charge the Plant.—The charge of ammonia should go into the low pressure part of the plant always. Figure 206 shows the proper manner of connecting the shipping drum. To charge, the expansion valve on the line should be opened wide and the valve on the drum should act for the period of charging as the expansion valve. If the evaporator is a brine cooler, the brine *must* be kept in circulation and the compressor and the condenser must be in operation. The first indication of the

emptying of the shipping drum is the appearance of frost on the drum and on the connections to the expansion valve, but the only way to be positive that the drum is empty is to weigh the drum and to compare this weight with the tare on the shipping tag. A sufficient number of drums of ammonia should be charged so that the plant will have enough refrigerant to operate.

The Amount of Ammonia Charge.—It is difficult to give exact rules for charging because of the variation in the amount of ammonia used in different cases. One rule,[2] which has been successful in practice, is as follows:

2-in. flooded atmospheric condenser, 12 pipes high, 20 ft. long, 140 lb. per stand.
2-in. standard atmospheric condenser, 24 pipes high, 20 ft. long, 60 lb. per stand.
$1\frac{1}{4}$ by 2-in. double pipe condenser, 8 pipes high, 20 ft. long, 20 lb. per stand.
$1\frac{1}{4}$ by 2-in. double pipe condenser, 10 pipes high, 20 ft. long, 25 lb. per stand.
$1\frac{1}{4}$ by 2-in. double pipe condenser, 12 pipes high, 20 ft. long, 30 lb. per stand.

The ammonia charge required for pipes submerged in brine or water is greater than that required for direct expansion in cold storage rooms. At $22\frac{1}{2}$ lb. per sq. in. suction pressure the following is recommended:

$1\frac{1}{4}$-in. pipe submerged, as in water coolers, brine coolers in ice tanks, $\frac{1}{7}$ lb. per lineal foot of piping.
$1\frac{1}{4}$-in. pipe exposed to the air, $\frac{1}{10}$ lb. of ammonia per lineal foot of pipe.
$1\frac{1}{2}$-in. pipe exposed to the air, $\frac{1}{6}$ lb. of ammonia per lineal foot of pipe.
2-in. pipe exposed to the air, $\frac{1}{3}$ lb. of ammonia per lineal foot of pipe.

It is also usual to figure on the receiver being one-half full.

If one knew definitely the quality of the refrigerant in each part of the system, the calculation of the amount of charge would not be difficult to get accurately for a certain operating set of conditions. The piping from the compressor to the condenser is operated usually with 100 deg. F. of superheat, or more, and from the brine cooler or the direct expansion piping to the compressor it is dry, saturated or superheated a few degrees. The evaporators are always flooded more or less, as is likewise the condenser, but all condensers (except the so-called flooded condenser) attempt to free the piping of the liquid condensate as soon as possible. The vertical and the horizontal shell and tube condenser is a type that frees itself quickly of the liquid. It must be remembered that the charge must be sufficient always to give a liquid seal on the outlet to the receiver, otherwise gas will pass through the expansion valve to the low pressure side of the system thereby choking the coils and reducing the capacity of the plant. There will also result a loss of ice on the return suction line, overheating of the piston rod, etc.

[2] J. Laichinger, National Association of Practical Refrigerating Engineers, 1923.

Usually the operator can also detect by the whistling flow through the expansion valve that gas is passing through, but the better method is to install a gage glass on the liquid receiver connected by means of automatic ball stop valves with the glass tubing. The operator then has an opportunity of observing at all times the amount of liquid seal on the outlet from the receiver. Best results, as regards the capacity of the compressor and of the evaporator can be obtained by frosting back to the compressor when operated at 25 or 30 lb. suction pressure or lower.

Other factors tending to affect the amount of charge required in the refrigerating plant are the pressures carried on the low pressure side and the rate of boiling of the refrigerant in the evaporator. The suction pressure affects the specific volume of the gas so that the higher the pressure the more weight is represented by each cubic foot of volume. Also, in operating, the nearer the temperature of the brine or the air in the direct expansion cold storage rooms is to the boiling temperature of the ammonia the greater the amount of flood it is possible to carry. In extreme conditions it will be found that sometimes "lost" ammonia is found in the coils where no evaporation is taking place on account of the room temperature being too low for evaporation to take place until the liquid *slops* over into the suction return line.

Operating the Expansion Valve.—The function of the expansion valve is to feed the evaporating surfaces with liquid. If the load is constant and the compressor is operating at a constant speed, conditions will automatically adjust themselves after a time with any setting of this valve, as the suction pressure will raise or lower until the weight being drawn into the compressor and condensed in the condenser is equal to the amount passing through the expansion valve.

However, each plant has its own correct operating pressure dependent on the duty to be performed. This is such an operating pressure as will give 3 to 5, 10 or even 15 deg. F. difference in temperature between the evaporating temperature of the refrigerant and the temperature of the fluid surrounding the surfaces. The low pressure piping is laid out with the idea in view of having this particular difference of temperature, so that each plant—to do its particular duty—must be operated for the conditions for which it was planned. Incidentally, it is generally understood that a liberal amount of evaporating surface, and a relatively high suction pressure will give best economy to the plant. With a certain amount of piping (if it is clean and operated in the correct manner), if the load cannot be carried, the only alternative is to *decrease* the suction pressure, but this lowers the capacity of the compressor and increases the horse power per ton of refrigeration and therefore such action should be considered the last resort.

Operation: Removal of Non-condensible Gases.—In starting up, even with a good vacuum previous to charging, there will be air in the system, and the operation of the compressor will result in bringing this air finally to the condenser where it will remain. If care is taken not to pump a vacuum on any part there will be no occasion for the amount of air or other gases to increase.[3] If care is taken not to pump a vacuum on any part of the plant there will be no occasion for the amount of air or other non-condensible gases to increase in amount. Ammonia or other refrigerants do not disintegrate under usual operating conditions, although some cheap lubricating oils will do so if the temperature of the discharge gases is excessive.

The action of the air in the condenser is to increase the pressure indicated by the condenser gage. This gage registers the total pressure, which is that exerted by the ammonia plus that due to the air present. The pressure of the air is that amount which it would itself exert if occupying the condenser volume by *itself* and at the same temperature. The pressure of the ammonia is that *surface tension* exerted by the ammonia during liquefaction and (if pure) is the pressure corresponding to the temperature of condensation. The law of partial pressures[4] is then

$$p_{\text{total pressure}} = p_{1\text{ of the air}} + p_{2\text{ of the ammonia}}$$

The "stuff" in the condenser is a mixture of a gas and a vapor, in different percentages in different parts of the condenser, but still a mixture. If air is present the film surrounding the condenser surfaces is nearly pure air while the condenser is in operation. The densest air is where the condensing water is coldest as a rule, but the non-condensible gases are naturally swept along in the direction of the flow of the ammonia. Purging, to free the condenser of foreign gases, should be at the point of greatest air density. This would be the upper part of the shell and tube condenser, and the bleeder condenser, but it is the lower part of the double pipe and the common atmospheric condenser.

Purging should be given considerable care. Without question a large amount of unnecessary purging is done and under no circumstances,

[3] Even sharp freezers requiring −20 deg. F. can secure this temperature without having a vacuum on the low pressure side of the compressor. The reason for air leakage into the compressor when operating under a vacuum is that the stuffing box on the rod will usually leak air *into* the cylinder. Pumping a vacuum on shutting down is very bad practice. Modern machines, even enclosed compressors, can take 150 lb. on the low pressure side without difficulty. It is advised on shutting down that the evaporator coils be pumped down to not lower than 5 lb. gage.

[4] See Chapter III.

whether a purging device is used or not, is it possible in *practice* to purge without losing ammonia. The easiest way to purge and yet conserve the ammonia is to "freeze" the ammonia out of the mixture by the use of refrigerating coils, made up either as a shell and coil or as a double pipe arrangement. All that is required is to cool the mixture to as low a temperature as practical, under which conditions the partial pressure of the ammonia will be reduced and the *weight of the ammonia per cubic foot* of mixture will be reduced in proportion to the density of the ammonia at that temperature. This method will not free the gas being purged out of the system of all the ammonia, and hence it is best to free the system of air and not permit it to get in by careless operation or by the unnecessary opening of the ammonia pipe lines for repairs. At the present time there are several patented devices to assist in purging, although the purge line extending into a bucket of water is the best arrangement for the small plant. Figure 207 shows the Hill non-condensible gas separator which is designed for automatic operation with a relief valve set at 100 lb. at the top of the shell.

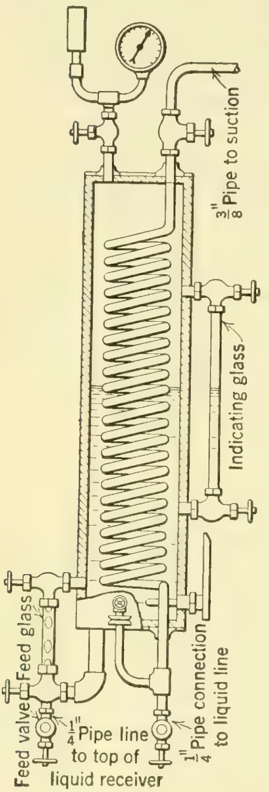


FIG. 207.—Purge Drum and Connections.

The Regenerator.—It is undoubtedly true that after the plant has been in operation for some time a certain amount of lubricant passes around the cycle continually. The oil separator does not separate all of the oil and the liquid in the liquid receiver does not separate like oil and water, but forms (to some extent) a solution of liquid ammonia and oil. The result is that oil will get into the low pressure piping, as well as water from the oil, from the ammonia, and from a number of other sources.

In order to get rid of the water and the oil the large plant uses a regenerator (Fig. 208) which operates on the principle of boiling out the ammonia at suction pressure. Warm water is usually used to keep the regenerator warm and connections are made to this device from all low points in the piping. The action of bleeding off should be done at frequent intervals; the ammonia returning at suction pressure and some-

what superheated back to the compressor. The residue consists of water, oil, solid materials and some ammonia.

Defrosting the Low Pressure Piping.—It is a mistake to believe that frost on the piping will give—on account of the larger diameter—a greater heat transfer than would be possible with bare pipe. Frost will decrease the heat transfer until finally the evaporator surfaces will fail to give results.

There are a number of ways of defrosting pipes:[5] The air in the room may be permitted to rise in temperature, or the refrigerant may be

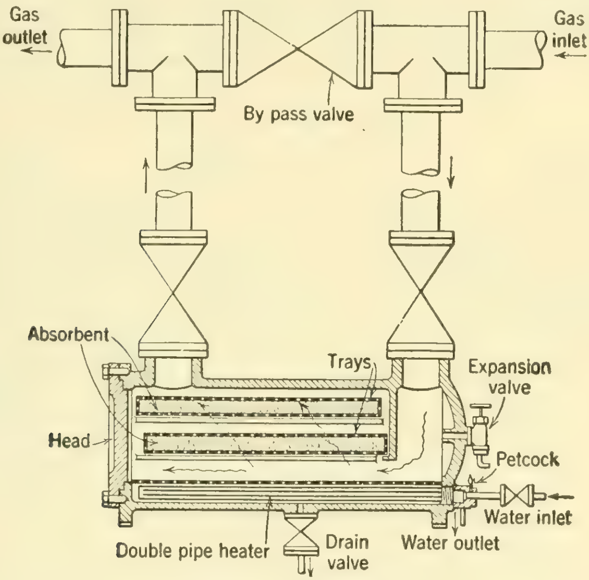


FIG. 208.—The Regenerator.

shut off or the machine stopped for a short time, the snow may be brushed off or the ice chipped off or defrosting may be accomplished by the use of hot gases from the compressor. In the last mentioned, the preferred method in some plants, the frosted pipes act as a condenser, and by the use of the by-pass valves, or by the use of a special compressor, the hot, superheated gas is pumped into the coils to be defrosted (Fig. 209). The larger plants make use of special defrosting lines which can be used at other times for the purpose of pumping out. Another advantage of the defrosting pipe line is that the warming of the piping permits the congealed oily mixture of mill scale, oxide, sand, etc., with the lubricator oil to flow to the traps at the low points of the

[5] See Chapter VIII.

piping. If traps are not installed, and it is desired to clean out the expansion piping, it is necessary to break into the piping, apply steam and then compressed air and finally blow out the line as at the time of erection. If there is any reason to believe that oil is congealed in parts of the condenser, it can be loosened by partly shutting off the water; but the better way would be to open up the two ends and then pass steam followed by compressed air through the piping, as just mentioned.

Lack of Capacity.—The refrigerating system is a closed one, using the refrigerant over and over, but the system is subject to troubles inherent in the manner of operation. Scum, dirt and scale may accumulate on the water side of the condenser due to impure water, frost and oil may cover the outside and the inside respectively of the evaporator

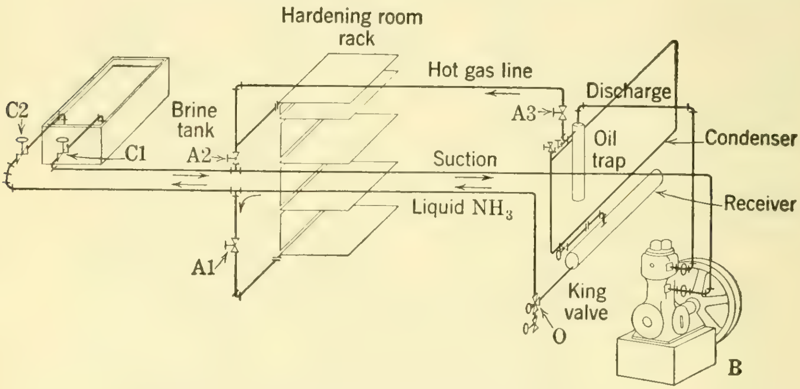


Fig. 209.—Defrosting Piping.

surfaces; leaky valves, piston rings or scored cyclinder walls may result in poor pumping ability in the compressor; obstructed circulation of the water or the ammonia may put the condenser out of commission or at least decrease its working ability; and faulty action in the brine cooler or the direct expansion piping may result in a poor heat transfer. An example of an improvement in operation is as follows:

a. Removal of a long accumulation of ice and frost permitted the machine to be operated at 15 lb. suction pressure gage instead of 3 lb. gage with equally good room temperatures.

b. Removal of a long accumulation of oil in the pipes permitted a further rise in the suction pressure from 15 to 17 lb. gage.

c. The piping had now been returned to normal conditions. This particular job was under-piped, and by an increase of the piping the proper room temperature was obtained by operating at 22 lb. gage.

Referring to curves showing the capacity of ammonia compressors under various operating conditions it will be seen in this case that a compressor (under 150 lb. condenser pressure) will require 9.2 cu. ft. per minute piston displacement per ton of refrigeration at 3 lb. gage and 3.7 cu. ft. at 22 lb. gage. With these two conditions the same compressor would give nearly 2.5 times the capacity at 22 lb. as at 3 lb. suction pressure.

Withdrawing the Charge.—Small and even moderate-sized plants are supposed to have receiver capacity, with what can be stored in the condenser or the evaporator coils, to store all of the ammonia charge in order that repairs can be made when needed. The larger plants cannot

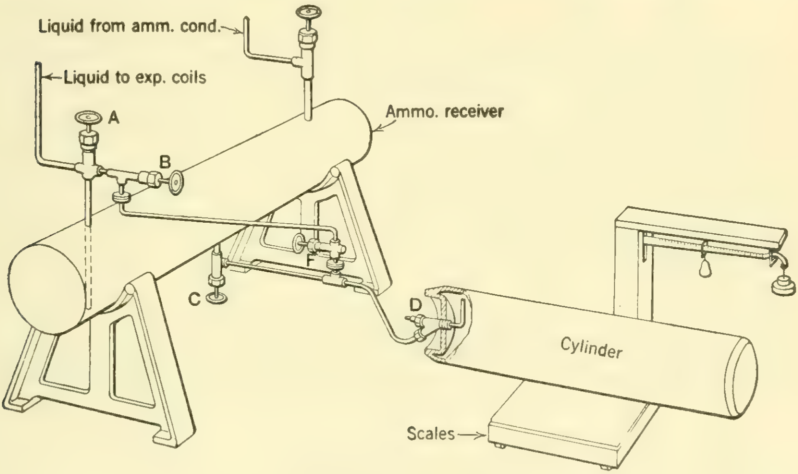


FIG. 210.—Withdrawing the Ammonia Charge.

do this, and withdrawal of part of the charge becomes necessary. In order to withdraw the ammonia an arrangement similar to Fig. 210 can be used. Note that in the figure the shipping drum has the bent tube pointing upwards so as to be able to free the drum of the air and the gas after it is partly filled with liquid. The method is to pump down the pressure by closing the valves A and C and opening B, F, and D. After the pressure in the drum has been lowered, close F and open C, repeating the process until the net weight in the drum is the amount the drum was designed for, usually 50 or 100 lb.

In withdrawing the ammonia charge from the system it is not necessary to use the complicated apparatus just mentioned. Usually it is sufficient to proceed as follows:

Connect an empty shipping drum to a point between the liquid

receiver and the expansion valve, with the valve on the shipping drum pointing down. Open the connections to the shipping drum, and cool the drum with cold water or ice. Note when the drum is full by weighing frequently. Particular care must be taken not to fill the drum beyond the safe amount indicated by the shipping tag.

Causes of Excess Pressure.—Excess pressure in the condenser may be caused by:

a. Too small a condenser, hot cooling water, or too small an amount of water for the temperature of the water used.

b. Inert gases in the condenser, mostly air or decomposed lubricating oil.

c. Poor heat transfer, due to dirt and scale from the cooling water, and oil and dirt on the ammonia side of the condenser.

d. Condensers partly filled with liquid ammonia due to an over charge (for the suction pressure being carried).

e. One or more stands not functioning due to broken or faulty valves so that the liquid cannot drain properly.

Causes of Loss of Capacity.—Loss of capacity in the plant may be caused by:

1. Insufficient piping surface, or frost on the outside and oil and scale on the inside.
2. Leaky valves, or safety head, piston rings or scored cylinders in the compressor; slow closing of the valves due to a number of causes.
3. Obstructions in the suction lines or broken valves preventing full opening.
4. Too small a charge of ammonia, thereby permitting gas to pass through the expansion valve.
5. The presence of water in the evaporating coils, the action of which is to require a lower pressure to be carried in order to obtain the temperature expected of anhydrous ammonia.

Figures 212 and 213 may be used as a convenient check on the capacity and other characteristics of small twin cylinder, single-acting vertical ammonia compressors for the rotative speeds indicated, which are nominal. For other speeds the results will be proportional. Tables 81 and 82 give details of valves, water and ammonia connections, etc., for the York enclosed compressor. These last details are very nearly standardized and are approximately true of all American manufacturers.

Starting Up.—The usual procedure in starting up the compressor is as follows:

1. Turn the water on the condenser and the compressor water jackets.
2. Start the brine pump, and (if a centrifugal pump) make sure that the brine is flowing.

3. See that the lubricators are in shape, and give the ammonia cylinder a small amount of cylinder oil by hand.

4. See that the suction and the discharge valves on the machine are closed, and open the by-pass from the discharge to the suction.

5. Start the compressor, and

6. Open the suction and the discharge valves. Close the by-pass valve.

7. Adjust the expansion valve so as to get proper operating conditions.

Testing for Leaks.—Ammonia losses[6] are greatest through the piston rod packing, valve stems, and flanged joints. It is next to impossible to keep the piston rod packing (Fig. 211) tight if the rod is not centrally located at all times in the stroke. Carelessness in taking up wear in the crosshead shoe is responsible for trouble of this sort. Also, the piston rod will wear in time more in the middle of the stroke than at the ends. Wear in excess of $\frac{1}{64}$ in. should be given attention, and probably will require turning down. It is wise to keep all valve stems well lubricated, and occasionally tighten on the packing. The sulphur stick is the best test for such leaks, a leak being indicated by white fumes, as already stated.

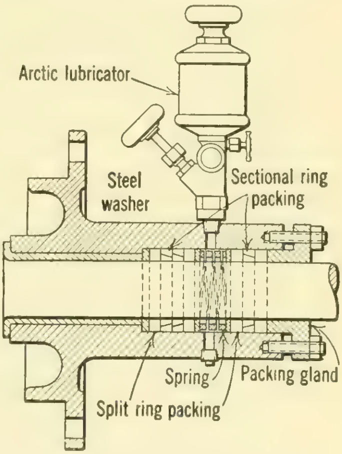


FIG. 211.—Piston Rod Packing.

The shell and tube and the atmospheric condenser can be tested with the sulphur stick while the outside is dry, and the pressure is up to about 200 lb. The double pipe condenser can be tested in the same manner, but the usual manner is to test the cooling water with Nessler's solution, which turns yellow (or, for heavy leaks, deep brown) when ammonia is present in the water. The same test is equally good for the brine cooler or the brine (ice) tank as well as the jacket water of the compressor. Nessler's solution can be made by dissolving 17 grains of mercuric chloride ($HgCl_2$) in 300 c.c. of distilled water and 35 grams of potassium iodide (KI) in 100 c.c. of distilled water. Add these, stir until a slight red precipitate is formed, and then add a cool solution of 120 grams of

[6] Average ammonia losses, according to H. T. Whyte, of the Consumers Ice Co., Chicago, cost 2.3c. per ton of ice, and Charles Neeson of the Knickerbocker Ice Co. New York, states that the average of 14 plants in Philadelphia is 2.0c. per ton.

potassium hydrate (KOH) in 200 c.c. of water and then dilute with water until 1000 c.c. is obtained. Add this to the $HgCl_2$ solution until

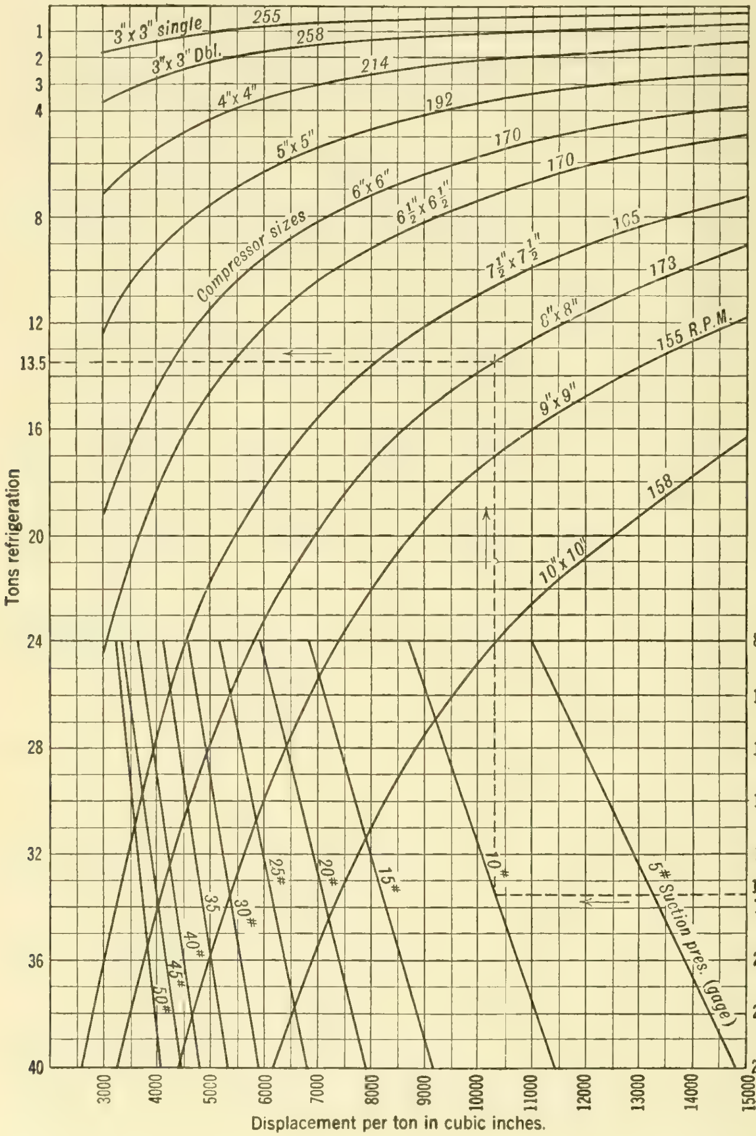


FIG. 212.—Capacity Curves—Enclosed Type Ammonia Compressors.

a precipitate is formed and then let it stand until it is settled and finally pour off the clear liquid into a blue bottle in order to prevent decomposition.

When testing calcium brine for ammonia it is necessary to precipitate the calcium out of solution first by adding to the sample of the brine

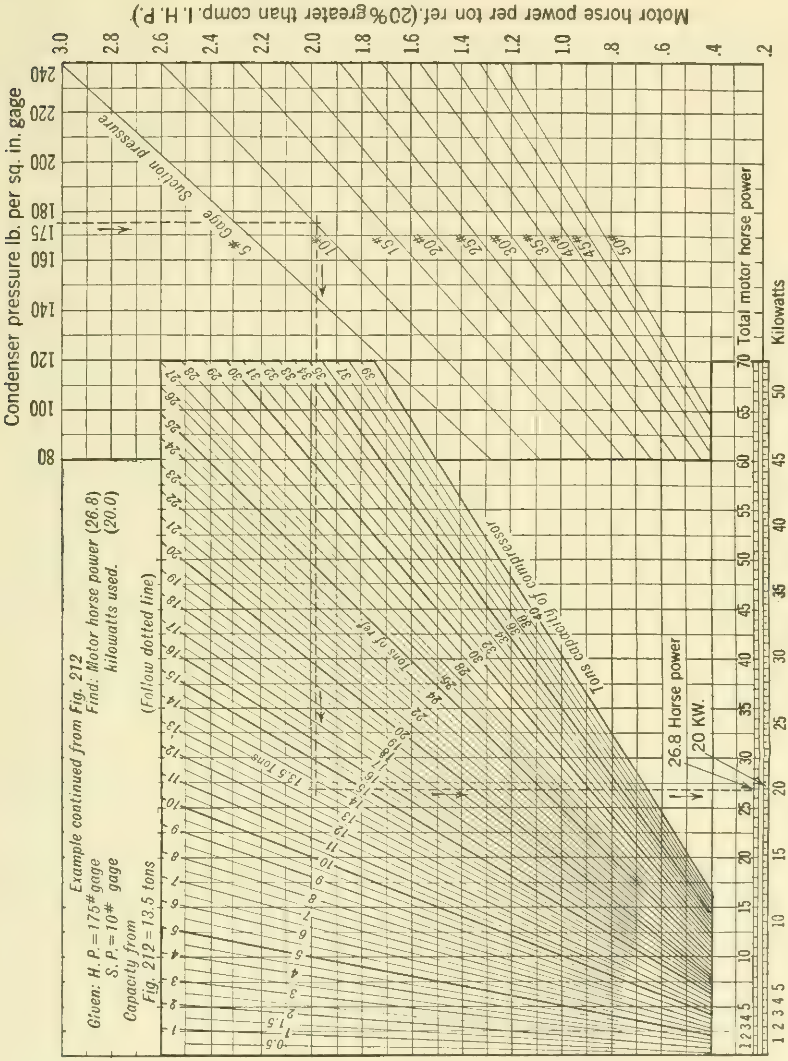


FIG. 213.—Power Requirements—Enclosed Type Ammonia Compressors.

diluted with water enough concentrated solution of sodium carbonate (Na_2CO_3) to precipitate out the calcium, and then add to the filtrate Nessler's solution. A change to a brown-colored solution indicates the presence of ammonia in the original solution.

Carbon Dioxide Systems.—In carbonic refrigeration the erection is practically the same as for ammonia. In charging the plant it has not been the custom to pump a vacuum as is always done in the case of ammonia, and in consequence there is considerable air in the piping after the operation of charging has been completed. Heretofore it has been considered satisfactory to rely on the numerous leaks in the piping to remove the air but it is quite evident that this custom will need to be abandoned and that provision will have to be made to pump out the air, at least partially, as in the case of systems with nominal condenser pressures.

Charging.—The amount of the charge in a carbonic system can be estimated in a manner similar to that in the ammonia plant by a calculation of the internal volume of the condensers, the piping on the high pressure and the low pressure sides and the amount of the liquid which should be in the condenser and the liquid lines. By the selection of the specific volume of the vapor for the pressures expected to be found in the pipes a fair estimate may be made of the required charge. The usual manner, however, has been the rather crude method of estimating the charge by the discharge pressure, which should approximate saturation for the temperature of the outgoing condenser water. If the condenser pressure is considerably lower than this amount then the charge is insufficient and more gas must be added. No gage glass is used on the liquid receiver of the carbonic plant, as such a glass would need to withstand pressures in excess of 1000 lb. per sq. in. nor has it been the custom to install liquid receivers of sufficient size to warrant the name.

The lack of a liquid receiver is surprising inasmuch as a pound of ammonia will produce as much refrigeration as 7 or 8 lb. of liquid carbon dioxide, and therefore the possibility of keeping the expansion valve sealed with liquid is decreased in proportion. For example, suppose that with ammonia it is usual to have a liquid receiver containing 100 lb. of liquid in order to supply the demands of the service imposed on it and still have a complete liquid seal at all times on the expansion valve. For the same variation of load there would be required some 700 lb. of carbon dioxide and it is positive that such a liberal charge is never supplied. Such being the case it is very likely that a considerable loss of capacity is frequent due to a loss of the liquid seal. The absence of a glass gage on the lower part of the condenser or the liquid receiver makes it necessary to rely on the pressures with the result that the careless operator permits the plant to run inefficiently. There is no question that this method of guessing at the proper operating conditions should be removed from carbonic refrigeration. To do this a suitable

liquid receiver should be installed and a liquid level indicator (for example a hollow steel ball within a brass tube, the location of which can be shown by means of a magnetised indicator.) There are also special small bore glass tubings manufactured at the present time which will stand from 1200 to 1400 lb. per sq. in. with safety and these can be used in liquid gage glasses if desired.

The action of *liquid* carbon dioxide in the refrigeration system is practically identical to that of ammonia and other volatile liquids where the liquid boils in the evaporating coils in absorbing heat and thereby becomes a vapor. The only disconcerting factor is the very low critical temperature (88.0 deg. F.) above which the gas cannot be liquefied even with excessive pressures. However this fact does not mean that no refrigeration is possible at condenser pressures greater than that corresponding to the critical temperature, but simply that the refrigerant flowing to the expansion valve is not a liquid although it is a very dense gas. The P–I diagram, shown in Chapter 7, indicates that when operating above the critical temperature a loss in the refrigerating effect per pound of carbon dioxide is inevitable but that otherwise there is no difference between carbonic and ammonia refrigeration.

In marine practice it is usual to overcharge the system during the passage through the tropics. The reason for this is that an overcharge, as the refrigerant does not condense, means a heavier discharge pressure which, for a stated temperature of the refrigerant at the expansion valve, means a smaller value of I at this point in the cycle. With the excess charge more work is performed on the gas during compression but the capacity is increased also. The actual pressure in the condenser under these conditions has no connection with the condensing water and it may be 1200, 1300 or 1400 lb. per sq. in. depending on the amount of the overcharge.

Great care must be exercised to operate with the proper sized opening of the expansion valve. If this pressure reducing valve is open too much the liquid seal is broken and partly cooled gas is permitted to pass into the evaporating coils. The effect of this action is an increased evaporating pressure and a reduced condenser pressure, the capacity of the plant becomes decreased and the desired temperatures in the evaporating coils will not be obtained. Charging more gas into the system will not give any satisfaction.

A more recent and advantageous method of operation, however, is the dual system, or the operation of the compressor by admitting two pressures into the cylinder. This is done by admitting the low pressure gas from the evaporating coils in the usual manner but at or near the end of the suction stroke ports in the cylinder are uncovered with

register with a suitable connection to the high pressure evaporating coils. When the piston overrides these ports in the cylinder the gas under the higher pressure flows in, the low pressure suction valve closes and the gas in the cylinder increases in pressure at constant volume. During the compression stroke the total weight of the gas in the cylinder is compressed, and this is considerably more than would have been the case if the low pressure suction gas only had been compressed. The manner of operation is as follows:

In the dual system a part of the refrigerant from the condenser which may be a liquid at 80 or 85 deg. F. or a dense gas at 90 or 100 deg. F. is passed through suitable cooling coils made cool by a fraction of this same carbon dioxide from the condenser which is regulated to absorb heat at a temperature of approximately 50 or 60 deg. F. The vapor resulting from the cooling of the refrigerant is at a higher pressure than that of the evaporating coils performing useful refrigeration and this gas enters the cylinder at the end of the suction stroke as already mentioned.

The advantage of the dual system is very evident. The refrigerant leaving the condenser may be at a high temperature and yet by the method just described it may be cooled to some temperature between 50 and 60 degrees. The same weight of refrigerant enters the evaporating coils in each case, but the amount of refrigeration per pound of the carbon dioxide is greater with the colder temperature at the expansion valve. The dual system requires more power for compression than the simple system.

Operating the Absorption Machine.—Referring to Chapter 3 it will be seen that a definite relationship exists between the temperature, concentration and the pressure of the aqueous solution in the generator. For proper operation at a particular steam pressure in the steam heating coils there must be a definite average concentration of the aqua in the generator. If the solution is too weak then capacity cannot be obtained without raising the steam pressure. Also the proper amount of aqueous solution must be circulated by means of the strong aqua pump (Table 15).

To charge the system a vacuum should be pumped on the entire system as in the compression plant, by using some form of vacuum air pump for the purpose. The refrigerant is charged in the form of aqua until the generator, absorber and exchanger have their proper amounts, then with anhydrous ammonia until the condenser, liquid receiver, evaporating coils and the return line to the absorber have the proper amount of ammonia for operating conditions. A calculation similar to that given for the compression plant can be made for the absorption plant.

To Operate.—In operation the following routine should be followed:

a. Turn on the water to the condenser, absorber, rectifier and weak aqua cooler.

b. Start the strong aqua pump at the rated speed, see that the weak aqua is flowing properly and that the weak aqua regulator is operating.

c. Start the brine pump, if there is one. Note the absorber pressure and consider whether it is the amount desired and to be expected with the temperature of water supplied to the absorber.

d. Turn the steam on the generator steam coils and adjust the pressure required by the design of the generator and the concentration carried. Note the amount of liquid in the liquid receiver and estimate whether the amount is sufficient.

e. Open and adjust the expansion valve.

In shutting down use the reverse order of operations.

Some disintegration occurs in the generator and inert gases are formed. To decrease this action a solution of potassium dichromate using 0.2 lb. per 100 lb. of aqua should be added to the generator.

TABLE 81

SUCTION AND DISCHARGE VALVES AS USED ON VERTICAL MACHINES

Size of Machine, Inches	Suction Valves			Discharge Valves		
	Number of valves	Diameter of opening, inches	Lift, inches	Number of valves	Diameter of opening, inches	Lift, inches
3×3	1	$1\frac{9}{16}$	$\frac{1}{4}$	1	3	$\frac{1}{8}$
4×4	1	$2\frac{5}{16}$	$\frac{3}{8}$	1	4	$\frac{3}{32}$
5×5	1	$3\frac{1}{8}$	$\frac{3}{8}$	1	5	$\frac{1}{8}$
6×6	1	$3\frac{5}{8}$	$\frac{5}{8}$	1	6	$\frac{7}{64}$
$6\frac{1}{2} \times 6\frac{1}{2}$	1	4	$\frac{5}{8}$	1	$6\frac{1}{2}$	$\frac{7}{64}$
$7\frac{1}{2} \times 7\frac{1}{2}$	1	5	$\frac{5}{8}$	1	$2\frac{3}{4}$	$\frac{3}{8}$
8×8	1	$5\frac{1}{4}$	$\frac{1}{2}$	1	$3\frac{1}{2}$	$\frac{3}{8}$
9×9	1	$6\frac{1}{8}$	$\frac{1}{2}$	1	$3\frac{1}{2}$	$\frac{3}{8}$
10×10	1	$6\frac{3}{4}$	$\frac{1}{2}$	3	$2\frac{1}{4}$	$\frac{3}{8}$

TABLE 82

York Y-15 Compressor Data

Size of Machine	Capacity in Tons	Speed in R.p.m.	Diameter of Shaft, Inches	Connections, Inches: Ammonia Low	Connections, Inches: Ammonia High	Connections, Inches: Water In	Connections, Inches: Water Out	By Pass, Inches	Weight Without Flywheel Pounds	Flywheel: Diameter, inches	Flywheel: Face, inches	Flywheel: Weight, pounds
3×3 *S*	$\frac{3}{4}$	255	$1\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{4}$	550	30	4	300
3×3 *D*	$1\frac{1}{2}$	258	$1\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{1}{4}$	760	30	4	300
4×4 *S*	$1\frac{1}{2}$	215	2	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{1}{4}$	640	32	$4\frac{1}{2}$	360
4×4 *D*	3	214	2	1	1	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{1}{4}$	990	32	$4\frac{1}{2}$	360
5×5 *D*	5	192	$2\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{8}$	1435	36	$6\frac{1}{2}$	600
6×6	8	170	3	$1\frac{1}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{8}$	1960	48	7	700
$6\frac{1}{2}$×$6\frac{1}{2}$	10	170	$3\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	1	1	$\frac{3}{8}$	2125	48	8	825
$7\frac{1}{2}$×$7\frac{1}{2}$	15	165	4	$1\frac{1}{2}$	$1\frac{1}{4}$	1	1	$\frac{3}{8}$	3000	50	10	1300
8×8	20	173	$4\frac{1}{2}$	2	$1\frac{1}{2}$	1	1	$\frac{3}{8}$	4300	60	10	1400
9×9	25	155	$5\frac{1}{2}$	2	$1\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$\frac{3}{8}$	6050	70	$10\frac{1}{2}$	1500
10×10	35	158	6	$2\frac{1}{2}$	2	$1\frac{1}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	8800	78	$12\frac{1}{2}$	2000

CHAPTER XI

THE TESTING OF REFRIGERATING PLANTS

The testing of refrigerating plants should not be a very different process, in principle, from that used for any other kind of plant except that the working medium is usually a noxious gas and is not permitted to leave the metal walls of the retaining pipe system or other apparatus. On account of this property of the refrigerant, and because the media employed are all quite expensive, due care must be taken to prevent accidents and undue loss. The testing consists in the measuring of the temperatures and pressures, the weights of volumes of the water, liquid refrigerant, and brine, the indicated horse power of the compressor, the revolutions per minute, and other quantities affecting the capacity of the plant.

In testing refrigerating plants the investigation may be for the entire plant, or it may be confined to a study of the compressor, the condenser, or the low pressure side. A study of the simple compressor would probably involve the action of the valves and ports, the degree of tightness of the piston rings and valves, and possibly the temperature of the discharge gas. In more complex machines, as in the case of dual compression or stage compression, the investigation could very well be amplified to include methods of varying the operating conditions and the economical methods of operation.

The condenser furnishes an excellent field for investigation at the present time. The important points are, as regards improving the effectiveness of the surface supplied, reduction of the amount of water required, and the reduction of the condenser pressure. The problem becomes one of securing values for heat transfer and of measuring the amount of water used and the refrigerant liquefied. At the present time (1927) little has been done on air-cooled condensers or on the automotive fan type of air-cooled radiators using water on the condenser.

The low-pressure side embraces methods of increasing heat transfer. This investigation requires a knowledge of the amount of the refrigerant boiled from the coils per unit of time and a complete knowledge of the temperatures involved. In most of the investigations on refrigerating systems the temperatures involved are the principal observations to be

made, and in the following the methods of securing temperatures will be given considerable prominence.

Temperature.—There are three ways that temperatures, such as are required by refrigerating engineers, may be measured: by the ordinary mercury in glass thermometer, by the electric resistance thermometer, and by the thermocouple. The mercury thermometer is the easiest to use for most work, but it is subject to considerable error. There is always a certain time lag in these readings, and they are of value only when the temperatures are reasonably constant. The greatest error is that due to conduction, especially in superheated gas temperatures. The graduations should be etched on the glass tube, and the thermometer should be calibrated for 32 deg. F. (the melting temperature of ice under standard atmospheric pressure) and possibly at 212 deg. F. (the boiling temperature of pure water under normal pressure). There is a tendency for the bulb to change its shape with time, a fact that affects all readings, and there is a possibility that the bore of the tube is not uniform in cross section, which would affect the intermediate readings.

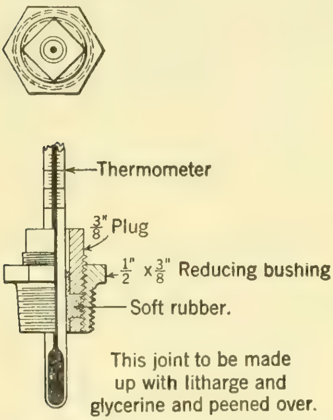


FIG. 214.—Thermometer Stuffing Box.

Temperatures requiring accuracy should be secured by means of calibrated thermometers throughout the range required in the test. In refrigeration the temperatures usually desired are between -20 and $+40$ deg. F. Frequently thermometers can be checked sufficiently well by calibration with a standard thermometer in a bath of oil, immersed up to the point of immersion of the test thermometer under test conditions.[1] The thermometer, for low temperature

[1] If the stem exposure correction must be applied, this may be calculated as follows:

$$\text{Correction} = 0.000085 \times N(T^\circ - t^\circ),$$

where

N = the number of degrees emergent from the bath;
T = the temperature of the bath;
t = the mean temperature of the emergent stem.

Example.—If the observed reading is 212 deg. F., the level of the bath (on the thermometer stem) is 100 deg. F. and the mean temperature of the air above the bath is 75 deg. F., then

$$\text{the correction} = 0.000085 \times 112 \times (212 - 75) = 1.3 \text{ deg. F.}$$

work, is best located as shown in Fig. 214, with the stuffing box and the bulb directly in the fluid the temperature of which is desired. The metallic thermometer well (Fig. 215), made of iron for ammonia and brass for other fluids, is a good arrangement for temperatures that do not require great accuracy. When temperatures are taken in this manner, care must be observed to lag the pipe sufficiently on both sides of the well so as to insure against radiation and other losses. It is wise to insert the thermometer in a tee (Fig. 215) so as to eliminate excessive radiation to the walls which may be, under certain conditions, as great as two or more degrees.

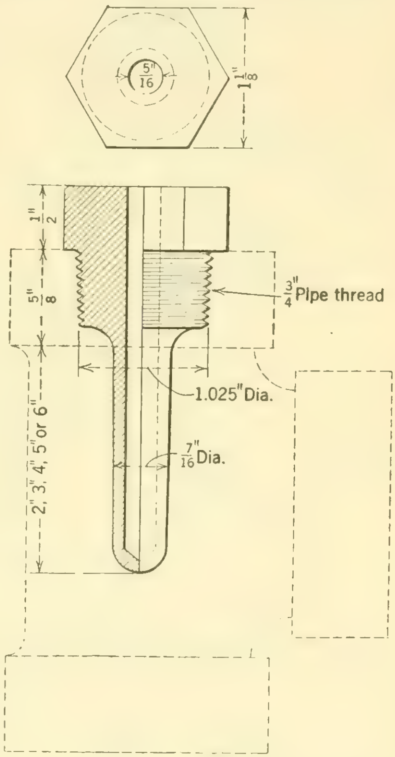


Fig. 215.—Thermometer Well in a Tee.

The Electrical Resistance Thermometer.—Resistance thermometers are based on the principle of the change of electrical resistance with the temperature, which means that nothing else—like a strain on the wire or the action of corrosion—should affect the resistance of the wire. For this reason these thermometers use platinum and nickel (the latter for moderate temperatures). In the construction a fine wire is wound on a spool, sometimes about $\frac{3}{8}$ in. diameter, and is sealed in a case.[2] The measurement of the resistance may be done by the use of a Wheatstone Bridge or by the use of a potentiometer, the latter being shown in Fig. 216.

The use of a potentiometer necessitates,[3] "two leads from each of the ends of the resistance, to which they should be soldered or welded. The resistance to be measured is represented by r_2, B is the battery to pass a current through r_2, R is a standard high-class resistance. The drop of voltage across R measures the amount of current passing through R. r_1 is the resistance which would be used to bring the current to a suitable value for the size of wire in r_2 and the value of $R + r_2$. The

[2] Bureau of Standards, Circulars No. S 407 and No. T 170.

[3] P. Nichols, Temperature Measurements, Refrigerating Engineering, 1924.

potentials, e and e', would be measured on the potentiometer and from Ohm's law,

$$r_2 = \frac{e'}{e}R.$$

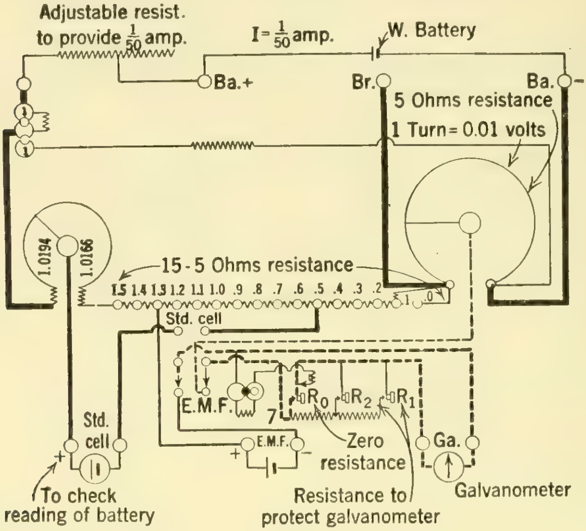


Fig. 216.—Type k Potentiometer.

"There are cases where the resistance principle can be usefully employed, such as where the resistance wire is not in the form of a bulb but is laid on a surface, or imbedded in a material when it is desired to obtain an average temperature over a considerable length or area. For this platinum would usually be too expensive and unnecessary. For short lengths nickel would be used, and for long lengths, copper. To be able to translate resistance values into temperatures, wire with equational relationship between the two would be purchased from an instrument maker, or a piece to be used would be calibrated. If this method is used, an analysis should be made of the readings which are obtained, in order to find the best value for r_2, and for the current passed through.

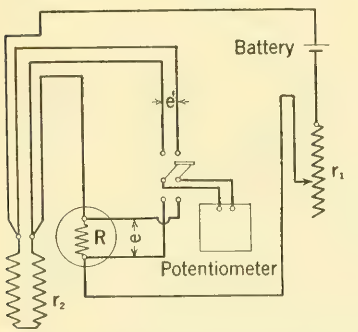


Fig. 217.—Arrangements for Electric Resistance Thermometer.

"The resistance and temperature coefficient—that is, its change of resistance per degree of temperature—of commercial copper is fully covered in B. of Std., S. 147 and S. 148. Commercial soft wire was found to have a fairly constant value in different samples and the temperature coefficient below 212 deg. F. has a linear relationship with temperature, so that

$$R_1 = R_0[1 + a(t_1 - t_0)].$$

where R_1 and R_0 are the resistances at t_1 and t_0, and a is a constant for a definite value of t_0. The symbol a represents the coefficient for constant mass; that is, we are dealing with a fixed weight of wire and not a fixed length, as the length will change with the temperature. Values for copper are:

t_0	a
32 deg. F.	0.00238
59 deg. F.	0.00223
68 deg. F.	0.00219
77 deg. F.	0.00214

The value for a within the range of 32 to 77 deg. F. also can be expressed as

$$a = 0.00255 - 0.0000053t.$$

"If, for example, the resistance of a piece of copper wire were measured at 59 degrees and found to be R, then the temperature at any other resistance, R_t, is found from the equation

$$t_1 = \frac{R_1 - R_0}{0.00223R_0} - t_0.$$

These data are useful under some circumstances, or for changes of temperature, as they only involve measuring the resistances of a piece of wire used at one known temperature. For accuracy, the value of R_0 and a should be obtained by finding the resistance of the actual pieces of wire to be used—preferably with the leads attached—at two temperatures.

"Nickel wire is more likely to vary, on account of impurities, and must be calibrated. The equation $\log R = a + bt$ holds approximately, and values for a and b can be determined by tests at two temperatures. For a good commercial wire $a = 0.0119$ and $b = 0.000372$ for t in degrees F. Table 83 will guide in fixing the size of the wire to be used. There is no advantage in too high a resistance, since although the difference in resistance will be larger, yet the sensitivity of the measuring instrument decreases. If the wire is uninsulated the apparent resistance will

be affected by moisture collecting on the wire. For accurate work all wire used should be calibrated."

TABLE 83

RESISTANCE PER FOOT OF WIRE AT 70 DEG. F.

Size of B. & S. Gage	Copper	Nickel	Size of B. & S. Gage	Copper	Nickel
21	0.013	0.079	31	0.130	0.811
22	0.016	0.100	32	0.164	1.004
23	0.020	0.125	33	0.207	1.275
24	0.025	0.159	34	0.260	1.619
25	0.032	0.200	35	0.328	2.047
26	0.041	0.254	36	0.414	2.572
27	0.051	0.314	37	0.522	3.183
28	0.065	0.405	38	0.658	4.019
29	0.082	0.503	39	0.830	5.270
30	0.103	0.643	40	1.047	7.144

Thermocouples.—The principle of the thermocouple is that of the production of a small electromotive force due to the contact of two

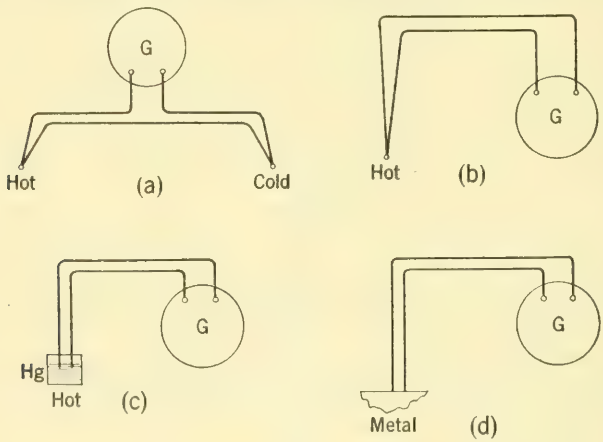


FIG. 218.—Simple Thermocouple Connections.

dissimilar metals which under certain well-defined conditions is a function of the difference of the temperature of the two junctions and of the metals used. A very simple thermocouple arrangement is illustrated in Fig. 218, in which a galvanometer is shown in the circuit, and in cases *b*, *c* and *d* the galvanometer is also the cold junction. In *a* the cold junction may be the temperature of the air in the room or it may be a

container with a mixture of ice and water. The wires must be insulated between junctions. Thermocouples may be used in series (Fig. 220), and the total e.m.f. will be the sum of each separate e.m.f., in which case the usual practice is to divide by the number of the couples in order to find the average, or it can be in parallel (Fig. 220). If a number of couples are in parallel, the resulting reading of the potentiometer will

TABLE 84

COPPER—CONSTANTAN THERMOCOUPLE

E.m.f.—Temperature, With Cold Junction in Ice
Deophysical Laboratory (L. A. Adams)

E.m.f. Millivolts	Temperature, Degrees F.	Temperature Difference	E.m.f. Millivolts	Temperature, Degrees F.	Temperature Difference
−1.0	−16.28	4.99	1.0	77.48	4.41
−0.9	−11.29	4.95	1.1	81.89	4.38
−0.8	− 6.34	4.91	1.2	86.27	4.36
−0.7	− 1.43	4.88	1.3	90.63	4.33
−0.6	3.45	4.84	1.4	94.96	4.32
−0.5	8.29	4.81	1.5	99.28	4.30
−0.4	13.10	4.77	1.6	103.58	4.28
−0.3	17.87	4.74	1.7	107.86	4.26
−0.2	22.61	4.71	1.8	112.13	4.23
−0.1	27.32	4.86	1.9	116.35	4.21
0.0	32.00	4.66	2.0	120.56	4.19
0.1	36.66	4.63	2.1	124.75	4.18
0.2	41.29	4.61	2.2	128.93	4.16
0.3	45.90	4.59	2.3	133.09	4.14
0.4	50.49	4.56	2.4	137.23	4.13
0.5	55.05	4.53	2.5	141.36	4.11
0.6	59.58	4.51	2.6	145.47	4.09
0.7	64.09	4.49	2.7	149.56	4.08
0.8	68.58	4.46	2.8	153.64	4.06
0.9	73.04	4.44	2.9	157.70	4.04
1.0	77.48		3.0	161.74	

give an average e.m.f. provided all of the leads to the common junction points are of equal resistance. At times it is very desirable to have two or more couples arranged in parallel. The factors which may affect the reading of the couple are as follows, but they are usually smaller than the accuracy of most investigations:[4]

"1. Lack of homogeneity of the wires. This will introduce no variation unless it occurs in a portion of the wire in which there is a temper-

[4] P. Nichols, Temperature Measurements, American Society of Refrigerating Engineers, May, 1924.

ature gradient, as exists somewhere between the junctions. The effect is equivalent in action to having thermo-elements in the wire itself, and with a difference of temperature there will be an e.m.f. generated. Such an effect will not be large and can be tested for by changing the depth of insertion of the couple wires in the hot medium, or by otherwise creating temperature gradients at different points. Such stray e.m.f.'s are also caused by strain in the wires, and sharp bends should be avoided in the larger sizes, especially where they emerge from the hot or cold media.

"2. Galvanic effects at any portion of the circuit. These will be reduced by orderly arrangement of the wires and by keeping them dry. It will be well to coat the insulation with paraffin.

"3. E.m.f. from contact of dissimilar metals in the lead system. This may be in switches or lead wires. Care in locating instruments or switches so that different portions are not at different temperatures, will reduce this.

"4. Caused by other electric circuits or by stray leakage. This can be tested for by opening and closing the switches in such lines to test as to whether the reading is different with or without those lines in use. Also by shielding the instruments—that is, mounting them on sheet metal plates, all of which are connected together.

"Too much importance cannot be paid to the proper insulating of the lead wires, terminal blocks and all parts of the circuits, especially if they are mixed up with high voltage used for heating or other test purposes. If the atmospheric conditions are good and the humidity low, there usually should not be any trouble; but in hot, humid weather, or damp working conditions, the stray leakage may become serious. The use of rubber covered copper leads is a good precaution and the ordinary No. 18 or 19 black fixture wire is very good in that it does not take on moisture at the surface so easily. The cotton covering over an ordinary flexible lamp cord may cause trouble, especially if the rubber is poor, and it is sometimes necessary to soak it in paraffin or cut it back for several inches at the ends, so as to expose the bare rubber.

"5. Corrosion and deterioration, with proper care of the couples, are not factors at low temperatures."[5]

The wiring diagram for the Type K Leeds and Northrup potentiometer is shown in Fig. 216. This includes the potentiometer, lamp and

[5] It should be noted that where the galvanometer or the deflection method is used the reading is a function of the resistance of the leads, and any change in the resistance such as is produced by resistance, corrosion, or the switches, may seriously affect the calibration, while with the potentiometer the reading is independent of the resistance of the circuit.

scale, selector switch, standard cell galvanometer, battery or dry cell, and the necessary thermocouples and cold junctions. The manner of procedure is to check the battery against the standard cell by getting such a resistance in the potentiometer line that the battery current will just neutralize that of the standard cell and the net result will be zero deflection of the galvanometer. The battery now, for a short interval of time, will deliver a standard e.m.f. which tends to make a current of electricity flow in a certain direction. The e.m.f. of the couple tends to cause a current in the opposite direction and the resistance of the circuit is varied until the two currents are equal and the galvanometer deflection is zero.

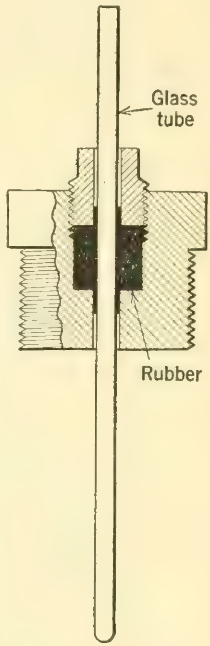


FIG. 219.—Thermo couple Fitting.

The Recording Thermometer.—The recording thermometer may be used at times when results of approximate correctness are desired. For remote indication of temperature there are only three distinct methods employed:

a. The direct expansion of liquids.
b. The vapor tension of volatile liquids.
c. The expansion of inert gases.

The recording thermometer requires a bulb to be placed at the point where the temperature is required, and also a fine bore flexible tube connection to the indicating gage device attached to clockwork.

Pressures.—Pressures are usually given in pounds per square inch above the atmosphere, and are indicated either by a tube mechanism or by a diaphragm.

The tube for ammonia is of special steel, hardened and tempered, whereas for brine, steam, etc., other materials can be used. The tube is elliptical in cross-section and more or less semicircular in shape. As pressure is applied the tube tends to straighten and, by means of links, a rack, and pinion, a dial is made to turn. No gage of this sort should be used without calibration by means of a suitable dead weight tester. For accurate work, calibration should be done before, after and (for long tests) during the test. The barometer should be recorded at all times when accurate data are required as the ordinary gage indicates the pressure above or below the atmosphere.

Besides the absolute pressures there is frequent need of pressure

differences. Sometimes this can be given by means of a U tube containing mercury, oil, water or other liquids. Figure 221 gives a good design for obtaining differential pressures, such as would be needed in the

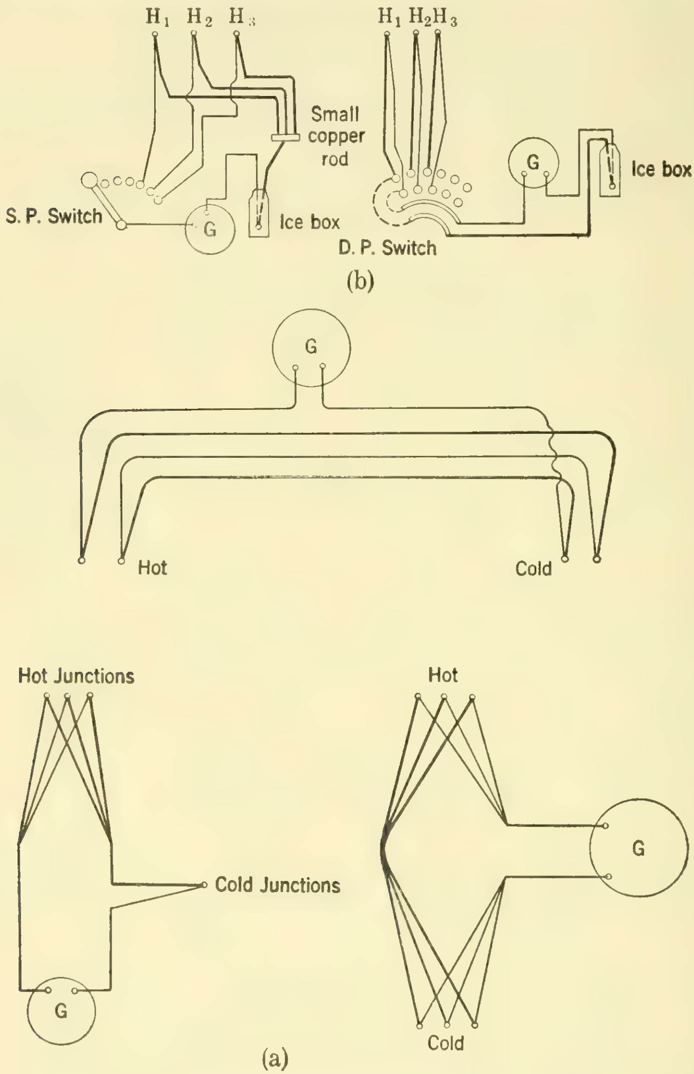


FIG. 220.—Thermocouple in Series and in Parallel.

ammonia gas line in order to show differences of pressure in the flow of superheated ammonia through pipes and fittings. The inverted U tube can be used in other cases, as, for example, in measuring the flow of

liquid ammonia through a thin plate orifice or Venturi meter; the differential height being in terms of the liquid flowing, which in this case is liquid ammonia. The float type differential gage,[6] designed for differential pressures up to 100 in. of mercury pressure, is shown in Fig. 222. This instrument can take static pressures up to 1000 lb. per sq. in.

Measurement of Quantity.—*Brine.*—If the brine is circulated by means of a centrifugal pump, then the Venturi and plate orifice method of measurement is satisfactory, but these methods are not accurate if

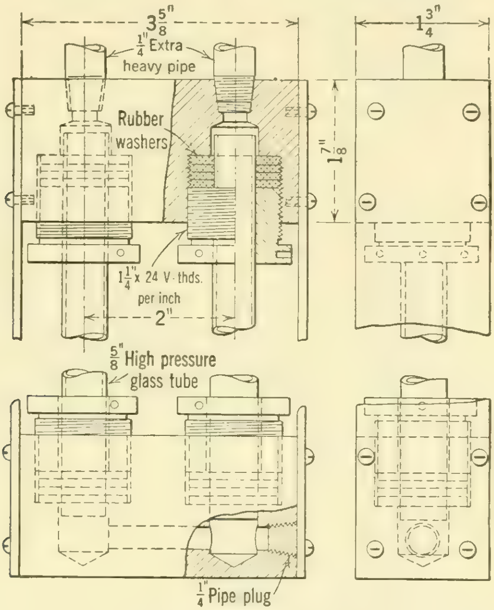


FIG. 221.—Differential U Tube Gage for Ammonia.

the reciprocating pump is used, on account of the pulsation developed. In any case the method of direct weighing on platform scales should be provided, or the orifice should be calibrated and then erected in place without change in the piping for a length equivalent to about 40 to 50 diameters of the pipe on both sides of the meter or orifice.

Water.—The amount of water passing over the condenser can be measured also by means of the thin plate orifice and the Venturi meter as well as the diaphragm piston and other forms of meters for measurement under pressure and by the rectangular and V notch weir. The principle of the fluid meter is as follows:

[6] The Foxboro Company.

Venturi Meter.

$$V_1 = A_1 \frac{1}{(R^4 - 1)^{\frac{1}{2}}} \times (2gh)^{\frac{1}{2}},$$

where

V_1 = the rate of flow in cubic feet per second;
C = a constant = 0.97 to 1.00 as a rule;

$$V_1 = M(2gh)^{\frac{1}{2}} \text{ theoretically;}$$

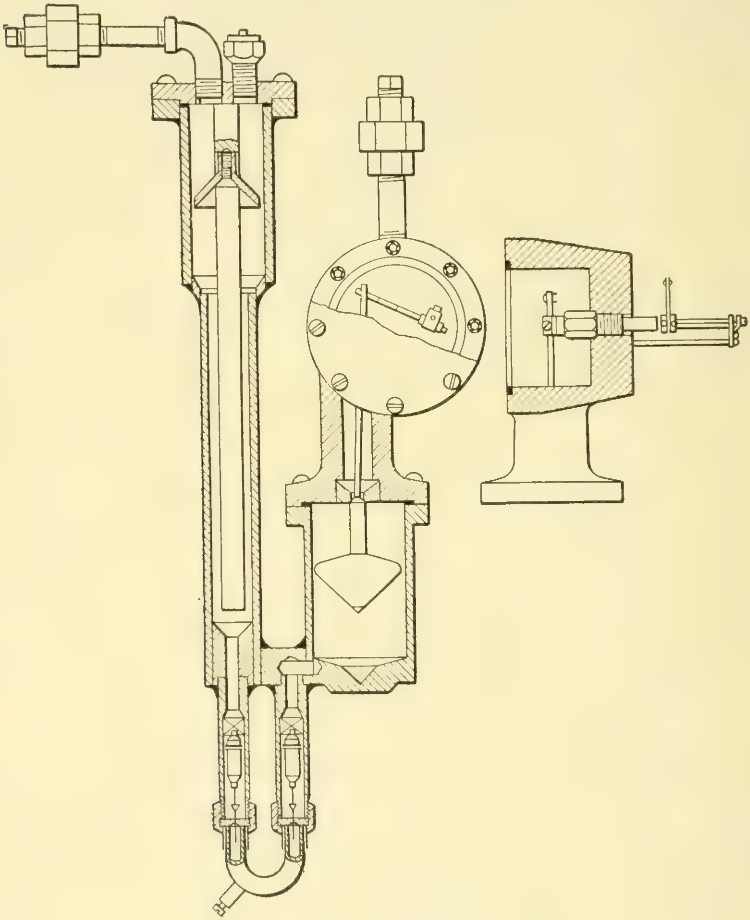


FIG. 222.—Differential Pressure Gage.

$$M = \frac{A_1}{(R^4 - 1)^{\frac{1}{2}}};$$

$$V_1 = CM(2gh)^{\frac{1}{2}} \text{ practically,}$$

A_1 = area entrance section at upstream connection, square feet.

$$R = \frac{D_1}{D_2},$$

where

D_1 = the entrance diameter;

D_2 = the throat diameter.

C is also given as the value of $f\left(\frac{D_2 \times S_2 \times d}{u}\right)$,

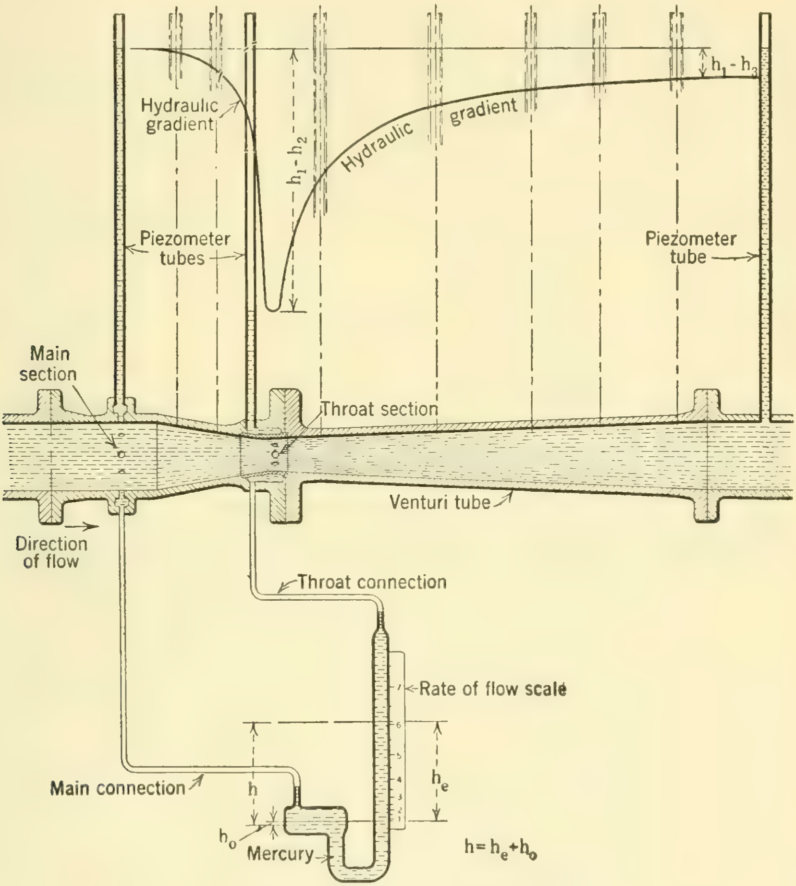


FIG. 223.—The Venturi Gage.

where

S = the speed;

d = the density;

u = the viscosity;

h is the differential level in feet of the liquid flowing.

The Thin Plate.—Here, also,

$$V = CM(2gh)^{\frac{1}{2}} \text{ cubic feet per second,}$$

where

$$M = \frac{A_1}{(R^4 - 1)^{\frac{1}{2}}};$$

A = cross-section pipe, square feet;

C = discharge coefficient = 0.60 to 0.62;

$R = \frac{D_1}{D_2}$, where D_1 = diameter of pipe, D_2 = diameter of orifice.

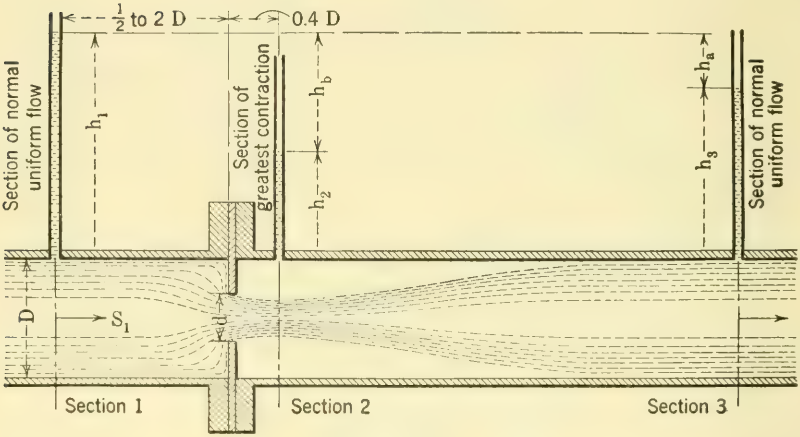


FIG. 224.—The Thin Plate Orifice.

If the differential pressure is not measured in feet of the liquid in the pipe but as a head h_0 (feet) of the liquid of density d_0, and d = density of liquid to be measured, then

$$V = CM\left(2gh_0\frac{d_0}{d}\right)^{\frac{1}{2}},$$

and

$$W = CM(2gh_0d_0d)^{\frac{1}{2}} \text{ pounds per second.}$$

Measurement of Refrigeration.—Of all the quantities to be observed in testing refrigerating plants that of measuring the amount of the refrigeration is the most important, as this quantity is valuable in determining the heat balance of the condenser and the brine cooler and the performance of the compressor. The volume and the weighing methods have been used extensively for this purpose, especially for ammonia.

The weighing method is a satisfactory means of obtaining the amount of liquid refrigerant passing through the expansion valve. The apparatus consists of two drums on platform scales connected by means of flexible pipes (Fig. 225) with the header from the condensers. Flexible pipes for equalizing must be installed also, and the connection from the condenser should be very liberal in cross-section and should be erected with a constant slope to the weighing drums.

If the knife edges on the platform scales are in good shape, the weight of the liquid refrigerant can be obtained, usually, to within $\frac{1}{2}$ per cent error. The usual procedure is to weigh the drum empty and full and to alternate from one drum to the other. A test should be made to see what weight on the platform will make the beam rise through its full distance. Ammonia cocks can be used to advantage here so as to

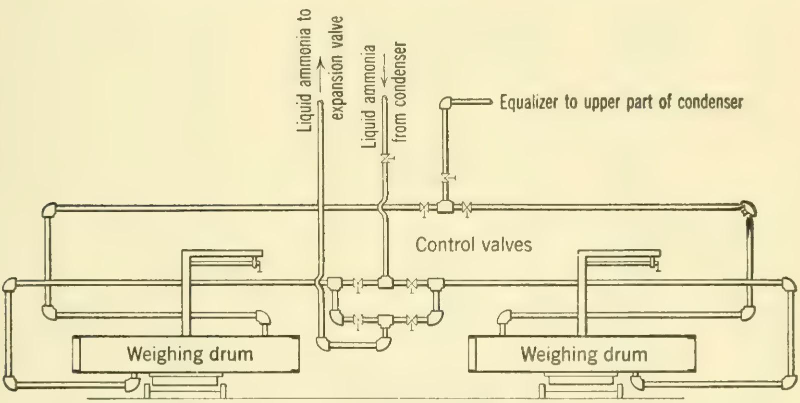


FIG. 225.—The Flexible Weighing Drum.

give a quick full opening and closing of the valve. Instead of weighing the total amount of ammonia passing through the expansion, the author prefers the method with the use of standard weights and an electric bell contact on the scale beam. In this case, with the drum normally empty, the standard weight is placed on the platform and the time noted with a stop watch when the beam makes contact at the top at which time the standard weight is removed. The time is again noted when the scale beam rings the bell again, as this will represent the lapse of time during the filling of the drum to an amount equal to the weight of the standard weight. Such a method requires *constant conditions* of operation, but no test is of any value unless the test conditions are constant. The test can be continued, alternating from one drum to the other, and the *rate of flow* can be secured in this manner. The total

weight recorded is not the *total* weight of condensate, but this total weight is not desired.

The Calibrated Tanks for Ammonia.—Where headroom will permit, the method of measuring the liquid ammonia by the use of calibrated

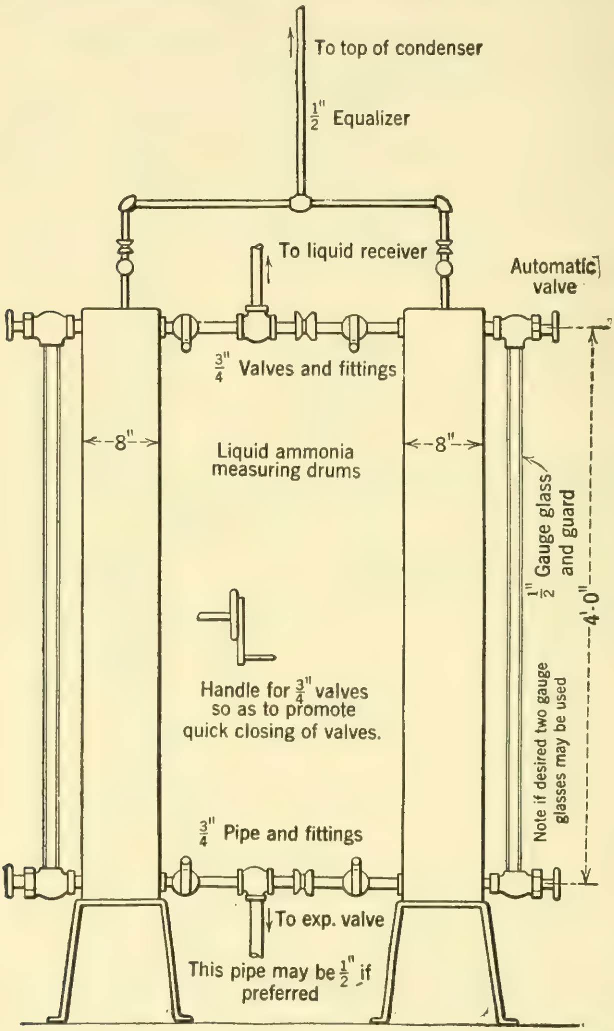


Fig. 226.—The Calibrated Drum.

tanks is sometimes better than by the use of the weighing drums (Fig. 226). In this case, also, cocks are better than valves if the cocks can be made tight, but tightness is the first essential. The upper connection

to the stand of condensers should be liberal, and each tank should be fitted with a gage glass of the best quality glass and with automatic stop valves. The calibration should be made after everything is piped ready for operation, by syphoning in water of a known weight and temperature, for which a scale of cubic feet can be constructed.

The particular advantage of the calibrated tank is that a check reading can be made readily. For example, operating conditions are established and maintained for a certain interval of time. After that it would be expected that the amount of liquid leaving one drum would always equal the amount entering the other. If this is not so then equilibrium has not been established, and the irregularity should be explained. As vertical tanks only are approved, a check can be made of the *rate of flow* by readings at frequent time intervals. As the tank calibration is in cubic feet the temperature of the liquid in the tank needs to be taken in order to reduce to pounds.

Both the weighing and the volume methods of measuring the liquid refrigerant need cumbersome apparatus, not easily moved from one plant to another like an engine indicator or a test wattmeter. Attempts have been made [7] to demonstrate the value of the Venturi meter for measuring liquid ammonia. While the Venturi meter and the flat plate orifice are excellent means of measuring water and brine, yet with liquid ammonia the case is different, for in this case there is a volatile fluid tending to form bubbles of gas with a decrease of pressure or an increase of temperature. For example, if the air temperature is 90 deg. F. and the condenser pressure is 150 lb. gage (corresponding to a temperature of saturation of 84.4 deg.), then there will be a tendency to absorb heat and vaporize. A drop of pressure will take place in the meter which will cause trouble also. The method advocated is to precool the liquid ammonia before it arrives at the Venturi (or thin plate orifice) by means of the coldest water available at the plant. The flow meter can be used to advantage to adjust the expansion valve, but it has not been proved to be a reliable means of securing the weight of the refrigerant.

Indicating the Power of the Compressor.—The valves of the ammonia compressor are not mechanically operated[8] and therefore they cannot be set improperly. The suction valve of the vertical compressor is usually a balanced valve, while the horizontal compressor can have a number of designs, and frequently employs the plate (ring plate) type. The indicator diagram is of no value in showing the setting of the valves, although it can show sluggish action, excess pressure in the cylinder due to inertia of the valve, excess pressure loss in getting the

[7] Compound Compressors, Geo. Horne, Refrigerating Engineering, 1923.

[8] The sleeve valve of the H. J. West compressor *is* mechanically operated (Fig. 47).

gas out of the cylinder and into the condenser and leaky valves and piston rings.

In using the steam engine indicator very great care must be exercised, especially if the speed is over 100 r.p.m., or inaccurate results will be experienced.[9] The principal sources of error are:

a. Inaccurate springs.

b. Excessive friction in the indicator on account of the piston.

c. Loss motion, poor design and excess inertia in the indicator mechanism.

d. Inaccurate reducing motion, inertia effects of the drum, stretching of the indicator cord.

The outside spring indicator has the advantage of the inside spring type because of the lesser temperature variation in the indicator cylinder, although this is less important in ammonia compression than in steam engine testing.

Reducing Motion.—An indicator card about $3\frac{1}{2}$ in. long is usually desired, and some connection is necessary to the crosshead or the shaft through a special mechanism in order to secure this reduced motion. If the crosshead is conveniently located some form of pantagraph is usually the best, or a specially designed reducing motion can be laid out on the drawing board for the particular compressor. Whatever type of reduction is used should be centered accurately, the pins should be of steel, hardened and ground, and arranged for lubrication. If the crosshead cannot be reached, an eccentric on the shaft or a crank and connecting rod should be used, designed for the *same* ratio of crank to connecting rod as in the compressor. The reduced motion can be extended to a convenient point, where attachments to the indicator can be made, by means of steel wire kept taut all the time by means of a helical spring.

For speeds of 200 r.p.m. and over the steam engine indicator is of doubtful value. Under such conditions some other form of indicator, the optical or the Bureau of Standards' diaphragm type, for example, should be used.

Other Test Observations.—There should be means of measuring the speed, the electrical power input (if electric driven), the density and the specific heat of the brine, and of the condensing water if this is not fresh water, the barometer, etc. If the compressor is steam-engine driven, indicator cards of the steam cylinder, the weight of the steam and the initial quality of the steam are desired. A test code for refrigerating machines has been prepared jointly by the American Societies of Mechanical and Refrigerating Engineers, and is given in the following:

[9] Experimental Engineering, Carpenter and Dieterichs, John Wiley and Sons.

Test Code for Refrigerating Systems

1. The Test Code for Refrigerating Systems is intended for use in the determination of the performance of compression systems in which compressors of the reciprocating type are used as well as absorption machines. The term "refrigerating system," as used in this Code, includes all the necessary machinery and apparatus directly used in the refrigerating cycle for determining the refrigerating effect, power consumption, and other primary data, indicated in the following tables.

It does not include ice making and other apparatus employed in the utilization of refrigeration. In so far as the fundamental operations of all the systems are in common, general rules will be laid down and where the systems differ separate rules will be given. For the test of the driving element in compression systems, the A.S.M.E. Power Test Codes for Steam Engines, Steam Turbines, Internal-combustion Engines, etc., should be consulted for details not contained in this Code.

Object

2. In accordance with the "Code on General Instructions," the object of the test should be first determined and recorded. If the object relates to the fulfilment of a contract guarantee, an agreement should be made between the interested parties concerning all matters about which disputes may arise (see "Code on General Instructions," Par. 6), and a full statement of the agreement entered in the report of the test.

Measurements

3. The principal measurements and quantities determined in a test are:
   (*a*) The quantity of the refrigerant circulated;
   (*b*) The various temperatures of the refrigerant in the cycle;
   (*c*) The various pressures of the refrigerant in the cycle;
   (*d*) The energy required to operate the driving element and the various auxiliary apparatus;
   (*e*) The various temperatures of condensing and cooling water;
   (*f*) The quanity of condensing and cooling water;
   (*g*) Quantity, quality and pressure of steam used in the steam cylinder or generator;
   (*h*) Principal dimensions of driving element (see appropriate code);
   (*i*) Quantity of composition and density of brine used;
   (*j*) Temperature of brine entering and leaving the evaporator;
   (*k*) Quality of liquor used in absorption plants.

Instruments and Apparatus

4. The instruments and apparatus necessary in carrying out the test are:
   (*a*) Suitable meter for measuring liquid refrigerant on the high- or intermediate-pressure side where it passes to the expansion coils of the brine cooler or elsewhere;
   (*b*) Platform scales and suitable tanks for measuring water and condensed steam;
   (*c*) Water meters, calibrated tanks or tank and platform scale for measuring, condensing and cooling water;

(*d*) Pressure gages and vacuum manometers, including a suitable U tube or mercury column to be connected at the low-pressure suction of the compressor. It is recommended that this be used in all cases rather than a pressure or vacuum gage for suction pressure under 20 lb. gage;
(*e*) Calibrated thermometers and thermometer mercury wells of sufficient lengths to extend beyond the centers of the pipes. Thermometers used for taking temperatures of brine shall be graduated in 1/10 deg. F. and readings estimated to 1/100 deg. F.;
(*f*) Barometer and psychrometer;
(*g*) Revolution counter or other accurate speed measuring device;
(*h*) Graduated stroke indicator for direct-acting pumps;
(*i*) Pycnometer, Mohr-Westphal balance, or calibrated hydrometer for specific gravity of brine;
(*j*) Calorimeter for determining the quality of the steam;
(*k*) Steam and ammonia indicator;
(*l*) Planimeter;
(*m*) Appropriate instruments for measuring power consumption of motor-driven machines.

The direction for the use, calibration and accuracy of the instruments and apparatus enumerated above are given in the various sections of the "Code on Instruments and Apparatus."

Preparation

5. Paragraphs 7 to 11 of the "Code on General Instructions" should be read, carefully studied and conformed to wherever they apply. The dimensions and the physical conditions of all parts of the plant essential to the object of the test should be accurately determined and carefully recorded.

6. The dimensions of the compressor and engine cylinder and the pump plungers or cylinders including the volume of the clearance should be accurately measured and recorded in accordance with Par. 6 of the A.S.M.E. "Test Code for Reciprocating Steam Engines," and Par. 12 of the A.S.M.E. "Test Code for Displacement Compressors and Blowers."

7. All measuring devices should be installed and in the case of the meters for measuring water or brine a means of calibrating by direct weighing or checking by calibrated tanks should be provided. The meter for measuring the liquid refrigerant shall be calibrated by means of a closed tank or receiver of known volume. This volume should preferably be determined by weighing the water which fills the tank or receiver. In the ordinary plant a liquid receiver may be used for this purpose.

8. The thermometer wells should be of steel for ammonia lines, and bronze for brine lines, and should extend into the pipe a sufficient distance so that the bulb of the thermometer will extend at least to the center of the pipe. When comparisons of temperatures are being made it is extremely important that the immersion in the well should be identical in all cases. These wells should be filled with mercury to the point indicated on the stem of the thermometer. Insulation on the cold lines should extend even to the top of the well, and the thermometers pass through corks fitted to the wells and bored to accommodate the thermometers. The thermometer well must be placed at least 10 ft. away from brine coolers or pumps so that the solution has uniform temperature where the readings are taken. Electric resistance

thermometers or thermocouples may be used in taking temperatures of brine as a check on mercury thermometers.

9. In testing a plant where the load is light a brine heater or a heater using waste condensing water flowing over the evaporating coils should be used. An agreement should be reached as to which of the interested parties is to supply and install the brine heater, if necessary, and all other apparatus required for the tests.

10. For ammonia a special steel indicator must be provided which has steel indicator cocks with short separate connections to the ends. The indicator should be provided with a stop so that a light spring may be used for studying suction conditions.

11. In ammonia absorption systems the anhydrous ammonia should not contain over 3 per cent moisture. Samples should be drawn before and immediately after the test and determinations of moisture made.

Operating Conditions

12. The operating conditions should conform to the object of the test and should prevail throughout the trial as pointed out in Par. 23 of the "Code on General Instructions." Uniformity of conditions and evenness of operation must prevail throughout the test. It is required under this Code that nothing but liquid shall enter the expansion valve and nothing but vapor shall enter the refrigerating machine. In commercial tests the allowable percentage of difference in the heat balance between the water and the refrigerant must be stated.

Starting and Stopping

13. The plant should be operated a sufficient length of time prior to the starting of the test to insure uniformity of conditions and, when these conditions are obtained, the test should be started and continued as stated in Par. 14 of this Code. It is essential to the accuracy of the test that all parts of the plant contain the same amount of heat estimated above some datum at the end as at the beginning of the test. To accomplish this all vessels containing liquid refrigerant should be supplied with suitable gage glasses so that the same quantity of fluid may be distributed alike at the beginning and end of the test and the quantities and temperature of the fluids in the various parts maintained as uniform as possible.

Duration

14. Each run shall be continued over a period of time not less than t hours, t to be calculated from the following formula:

$$tw_a = 3w_b$$

where w_a = approximate weight of anhydrous refrigerant circulated in pounds per hour, and

w_b = total weight of anhydrous refrigerant contained in the system.

Records

15. The data should be taken and recorded in the manner prescribed in the "Code on General Instructions," Pars. 24 to 35. Fifteen minute readings will be sufficient, except where there is considerable fluctuation in the readings, in which case more frequent readings must be taken to insure good averages.

16. Each indicator card should be marked with the date, time of day, cylinder and end of cylinder, and identification mark and strength of indicator spring, as pointed out in the "Test Code for Reciprocating Steam Engines," Par. 20. To assist in the uniformity of operation a chart should be plotted during the test of the principal quantities, including the estimated tonnage, as pointed out in Par. 39 of the "General Instructions."

17. The data and results should be tabulated in accordance with the form shown in the following tables, adding items not provided for and omitting items not needed to conform to the object in view. Unless otherwise indicated the items refer to the numerical readings which are recorded in the log.

Calculation of Results

18. *Standard Ton of Refrigeration.*—The now generally adopted recommendations of the Joint Committee of The American Society of Mechanical Engineers and American Society of Refrigerating Engineers on Standard Tonnage Basis for Refrigeration are as follows:

(1) A standard ton of refrigeration is 288,000 B.t.u.

(2) The standard commercial ton of refrigeration is at the rate of 200 B.t.u. per minute.

(3) The standard rating of a refrigerating machine,[1] using liquefiable vapor, is the number of standard commercial tons of refrigeration it performs under adopted refrigerant pressures.[2]

[1] Note 1. A refrigerating machine is the compressor cylinder of the compression refrigerating system, or the absorber, liquor pump, and generator of the absorption refrigerating system.

Note 2. Nothing but liquid shall enter the expansion valve, and nothing but vapor shall enter the refrigerating machine.

Note 3. There shall be 9 deg. F. (5 deg. C.) sub-cooling of the liquid entering the expansion valve and 9 deg. F. (5 deg. C.) superheating of the vapor entering the refrigerating machine, the points at which the sub-cooling and the superheating are determined to be within 10 ft. of the expansion valve and refrigerating machine, respectively.

[2] Note 4. The inlet pressure is that which corresponds to a saturation temperature of 5 deg. F. (−15 deg. C.).

Note 5. The outlet pressure is that which corresponds to a saturation temperature of 86 deg. F. (30 deg. C.).

Note 6. These pressures are measured outside and within 10 ft. of the refrigerating machine, distances which are measured along the inlet and outlet pipes, respectively.

19. The refrigerating output of the system shall be expressed in standard commercial tons of refrigeration as calculated from the weight of liquid refrigerant and the amount of available cooling effect produced in the evaporator. This method shall be employed in the case of all refrigerants where the physical properties have been determined and where tables of such properties are recognized as sufficiently accurate. The tables of thermodynamic properties employed shall be those adopted and published by The American Society of Refrigerating Engineers.

20. The weight of refrigerant circulated shall be determined by a suitable liquid meter as prescribed in the A.S.M.E. Code on Instruments and Apparatus. As an

alternate method the weight of liquid refrigerant may be calculated from the volume as obtained from a pair of calibrated receivers or by direct weighing.

21. The amount of available refrigerating effect shall be the weight of refrigerant circulated per hour multiplied by the difference in heat content on entering and leaving the evaporator. Where there is more than one evaporator in which different pressures exist, the weight of refrigerant to each evaporator and the total heat from the vapor must be determined separately and the total refrigerating effect may then be computed by the following formula:

$$Q_e = w_a(h_1 - h_4)$$

where Q_e = total refrigerating effect per hour;
w_a = weight of anhydrous refrigerant circulated in pounds per hour;
h_1 = total heat of the vapor leaving the evaporator;
h_4 = total heat of the liquid at the expansion valve.

22. The refrigeration effect expressed in standard commercial tons of refrigeration is

$$R.E. = \frac{Q_e}{12{,}000}$$

23. *Indicated Horse Power.*—The indicated horse power for each end of the cylinder is found by using the formula:

$$\text{i.hp.} = \frac{p_m LAN}{33{,}000},$$

where p_m = mean effective pressure in pounds per square inch;
L = length of the stroke in feet;
A = area in square inches of the piston less the area of the piston rod, if any;
N = number of revolutions per minute.

The total horse power of the cylinder is the sum of the horse power developed in the two ends.

24. *Mean Effective Pressure.*—The mean effective pressure should be found by dividing the area of the indicator diagram in square inches as determined with a planimeter by the length of the diagram in inches, and multiplying the quotient by the scale of the indicator spring. If a planimeter is not available, the approximate mean effective pressure may be determined by finding the average height of the diagram in inches as obtained by averaging a suitable number of equally spaced ordinates, at least twenty, measured between the lines of the forward and return strokes, and then multiplying this average by the scale of the spring. The length of the indicator diagram is the measured distance along the atmospheric-pressure line between ordinates erected perpendicular to it and passing through the ends of the indicator diagram.

25. *Theoretical Horse Power.*—The theoretical horse power required to compass adiabatically the quantity of vapor shall be found in the case of ammonia or fluids having standard tables or charts, by referring to the Mollier Chart or the tables adopted by The American Society of Refrigerating Engineers.

$$hp = \frac{(h_1 - h_2) \times w_a \times 778}{33{,}000 \times 60},$$

where

h_1 = heat content of vapor at compressor inlet;
h_2 = heat content of vapor after adiabatic compression;
w_a = weight of fluid circulated, pounds per hour.

26. In the case of fluids not having published charts or tables, the following method may be used to calculate the theoretical horse power required to compress the vapor adiabatically at the average rate for a single stage or for any stage of mutiple-stage compressors:

$$hp = \frac{144k}{33{,}000(k-1)} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right],$$

where v_1 = volume compressed, cubic feet per minute;
p_1 = absolute intake pressure, pounds per square inch;
p_2 = absolute discharge pressure, pounds per square inch;
k = the exponent in pv^k = constant.

27. *Gross Horse Power.*—The gross horse power is the indicated horse power in the steam of power cylinders in the case of a steam or internal combustion engine-drive compressor; the electrical horse power multiplied by the motor efficiency in the case of a direct connected motor-driven compressor; and the brake horse power delivered to the compressor shaft in the case of a belt-driven or gear-driven compressor. If motor efficiency determined under the A.I.E.E. rules is used, the value obtained will be a conventional one, and will always be higher than the actual efficiency of the motor. An agreement must therefore be made by those interested in the test as to the tolerance to be applied to the "conventional efficiency" to convert the figure into actual efficiency.

28. *Electrical Horse Power.*—The electrical horse power of a motor is found by dividing the input at the terminals expressed in kilowatts by the constant 0.746. In the case of an alternating-current motor, the input determined, whether expressed in electrical horse power or kilowatts, should be the total input. When the power for excitation is taken directly from the compressor shaft, the total input is that indicated at the a.c. motor terminals. When the power for excitation is furnished from an independent source the total electrical horse power used in calculating the gross horse power (Par. 27) is that indicated as the motor terminals plus an equivalent power required to run the exciter.

29. *Volumetric Efficiency.*—The volumetric efficiency is the ratio of the capacity of the compressor to displacement. The capacity is the actual amount of vapor compressed and delivered, expressed in cubic feet per minute at intake temperature and pressure.

30. *Mechanical Efficiency of the Compressor.*—The mechanical efficiency is the ratio of the indicated horse power of the compressor cylinder to the indicated horse power in the power cylinders in the case of a steam-driven or internal combustion engine-driven compressor, and to the brake horse power delivered to the shaft in the case of a power-driven machine..

31. *Compressor Efficiency.*—The compression efficiency in a single-stage compressor is the ratio of the horse power required to compress adiabatically all the vapor delivered by the compressor to the horse power developed in the compressor cylinder, as shown by the indicator cards. The compression efficiency for any par-

ticular cylinder of a multi-stage compressor is the ratio of the horse power required to compress adiabatically all of the vapor delivered by the compressor through the observed pressure range for the particular cylinder in question to the horse power developed in the particular cylinder in question as shown by the indicator cards. The two factors involved in this ratio are defined in Pars. 23 and 25, respectively.

32. *Condenser Performance.*—The total heat removed by the condensing water shall be calculated from the observed data when the enclosed type of condensers, either shell and tube or double pipe, are used. The total heat may also be calculated from the quantity of refrigerant circulated. In this case

$$Q_e = w_a(h_3 - h_4),$$

where Q_e = heat removed per hour;
h_3 = heat content of vapor entering condenser;
h_4 = heat content of fluid leaving condenser;
w_a = weight of refrigerant condensed per hour.

33. *Determination of Heat Transfer in Condensers.*—The logarithmic mean difference is based upon a constant temperature of liquefaction throughout the condenser on the assumption that the heat transfer is uniform throughout the condenser. While the calculation of heat transfer per square foot of refrigerant surface per hour per degree of logarithmic mean difference does not give entirely satisfactory results, the determination of heat transfer shall be calculated in this manner.

$$\text{Log. mean difference} = \frac{t_2 - t_1}{\log_e \dfrac{t_e - t_1}{t_e - t_2}},$$

where t_e = temperature of liquefaction corresponding to condenser pressure;
t_1 = temperature incoming water;
t_2 = temperature outgoing water.

If a complete condenser test is to be made the A.S.M.E. Test Code for Condensing Apparatus and the Code on Instruments and Apparatus should be carefully studied.

34. *Evaporator Performance.*—The evaporator performance shall be calculated from the observed data. In this case

$$Q_e = w_a(h_5 - h_6),$$

where Q_e = heat removed per hour;
h_5 = heat content of vapor, leaving evaporator;
h_6 = heat content of liquid entering expansion valve;
w_e = weight of refrigerant entering evaporator in pounds per hour.

35. *Determination of Heat Transfer in Evaporator.* (*a*) Brine Coolers.—Brine cooler performance shall be expressed in terms of heat transfer per square foot of cooling surface per hour per degree of logarithmic mean difference.

$$\text{Log. mean difference} = \frac{t_1 - t_3}{\log_e \dfrac{t_1 - t_5}{t_3 - t_5}},$$

where t_3 = temperature of brine out of evaporator;
t_4 = temperature of brine to evaporator;
t_5 = temperature of vaporization corresponding to evaporator pressure.

(*b*) The performance of ice tanks and all other forms of expansion coils or apparatus will not be specifically covered in this Code, which is intended to include the main items of the refrigerating cycle rather than the products of the cycle.

36. *Water Horse Power.*—The water horse power output at observed total suction and discharge pressure is

$$\text{w.hp.} = \frac{(\text{lb. of liquid per minute}) \times (\text{total head in ft.})}{33{,}000}.$$

The gross or brake horse power input shall be observed for all pump-driving elements and the efficiencies calculated.

37. *Heat Balance.*—The general formulas for heat balance are as follows:

$$\text{Compression Cycle: } Q_e + Q_w = Q_1 + Q_3;$$

$$\text{Absorption Cycle: } Q_e + Q_s = Q_1 + Q_2 + Q_3,$$

where Q_e = heat absorbed by evaporating refrigerant;
Q_w = heat equivalent of work in compressor;
Q_s = heat imparted by steam in generator;
Q_1 = heat rejected in condenser;
Q_2 = heat rejected in absorber;
Q_3 = heat rejected or radiated in addition to Q_1 and Q_2.

38. For purposes of illustration the following list of quantities involved in the computation of the heat balances of compound compression systems is given:

Heat Absorbed (B.t.u. per hour)

(*a*) Heat absorbed in evaporator;
(*b*) Heat entering evaporator insulation;
(*c*) Heat absorbed in low-pressure suction main;
(*d*) Heat absorbed in low-pressure suction trap;
(*e*) Heat equivalent of work done in compressor;
(*f*) Heat absorbed from engine room through cold surface of low-pressure compressor;
(*g*) Heat absorbed through surface of intermediate liquid receiver;
(*h*) Heat absorbed through surface of intermediate-pressure liquid line;
(*i*) Heat absorbed from engine room through surface of high-pressure suction main;
(*j*) Heat absorbed through cold surfaces of high-pressure compressor;
(*k*) Heat absorbed or rejected through condenser shells.
(*l*) Heat absorbed or rejected through receivers.
(*m*) Heat absorbed or rejected in high-pressure liquid line.

Heat Rejected (B.t.u. per hour)

(*n*) Heat rejected by hot surface of low-pressure compressor;
(*o*) Heat rejected from low-pressure discharge main between low-pressure compressor and intermediate vapor cooler;
(*p*) Heat rejected in intermediate vapor cooler;
(*q*) Heat rejected to engine room by intermediate vapor cooler;
(*r*) Heat rejected in discharge main from intermediate vapor cooler to intermediate liquid receiver;
(*s*) Heat rejected by hot surfaces of high-pressure compressor;
(*t*) Heat rejected by high-pressure discharge main and oil separator between machine and condensers;
(*u*) Heat rejected in ammonia condensers;
(*v*) Heat rejected in liquid cooler;
(*k*) Heat rejected or absorbed through condenser shells;
(*l*) Heat rejected or absorbed through receivers;
(*m*) Heat rejected or absorbed in high-pressure liquid line.

Before concluding tests, and making a final report, it is recommended that Par. 20 of the "Code on General Instructions" should be carefully read. The data and results should be reported in accordance with the forms prescribed by the committee.

CHAPTER XII

PIPING

A large amount of the equipment in the compression, and very nearly all of it in the absorption, refrigerating plant is *piping*. In some special cases, as, for example, the plant using the shell and tube condenser, the shell and tube brine cooler and, in the extreme case, a brine spray system of cooling air, the amount of piping would be comparatively small. In most cases, however, both the suction and lines should be calculated carefully, as well as the condenser headers and liquid drains and the piping on the low pressure side, or the brine piping if brine is used. Like all other branches of engineering, the piping must be efficient as regards operation and overhead costs, it must be arranged for draining, cleaning, pumping out and repairs, and it must be safe and self-contained. These considerations will be taken up in the following chapter.

Kind of Pipe.—Full weight steel pipe is quite generally used for ammonia at the present time, even for condensers and ice tanks, whereas carbon dioxide refrigeration uses extra heavy steel pipe, because of the much heavier unit pressures. Wrought-iron pipe is not so commonly used as it was some years ago, because it has been found that steel pipe can be depended on much better than formerly, both as regards the welded joint and the thickness of the walls. It is usual to specify butt-welded pipe for sizes 2 in. and smaller, but lap-welded pipe (drawn down from a larger size) for normal diameters larger than 2 in.

Piping at the Compressor.—The headers and connections at the compressors are usually irregular in design and therefore are frequently welded, although the cast iron or cast semi-steel bends are used whenever the cost of the casting is less than the welded header. Considerable care should be taken with the piping to eliminate changes in the direction of the gas flow as far as possible, since it has been determined that irregular flow through the cylinder ports (with sharp changes of direction of flow) and the suction and discharge bends is much more likely to cause loss of pressure of the gas than is the high velocity of the gas flowing through the valves due to small cross-sectional area. The discharge port areas sometimes are designed for 10,000 ft. per min. gas velocity, but if this is so the ports should be straight and when changes

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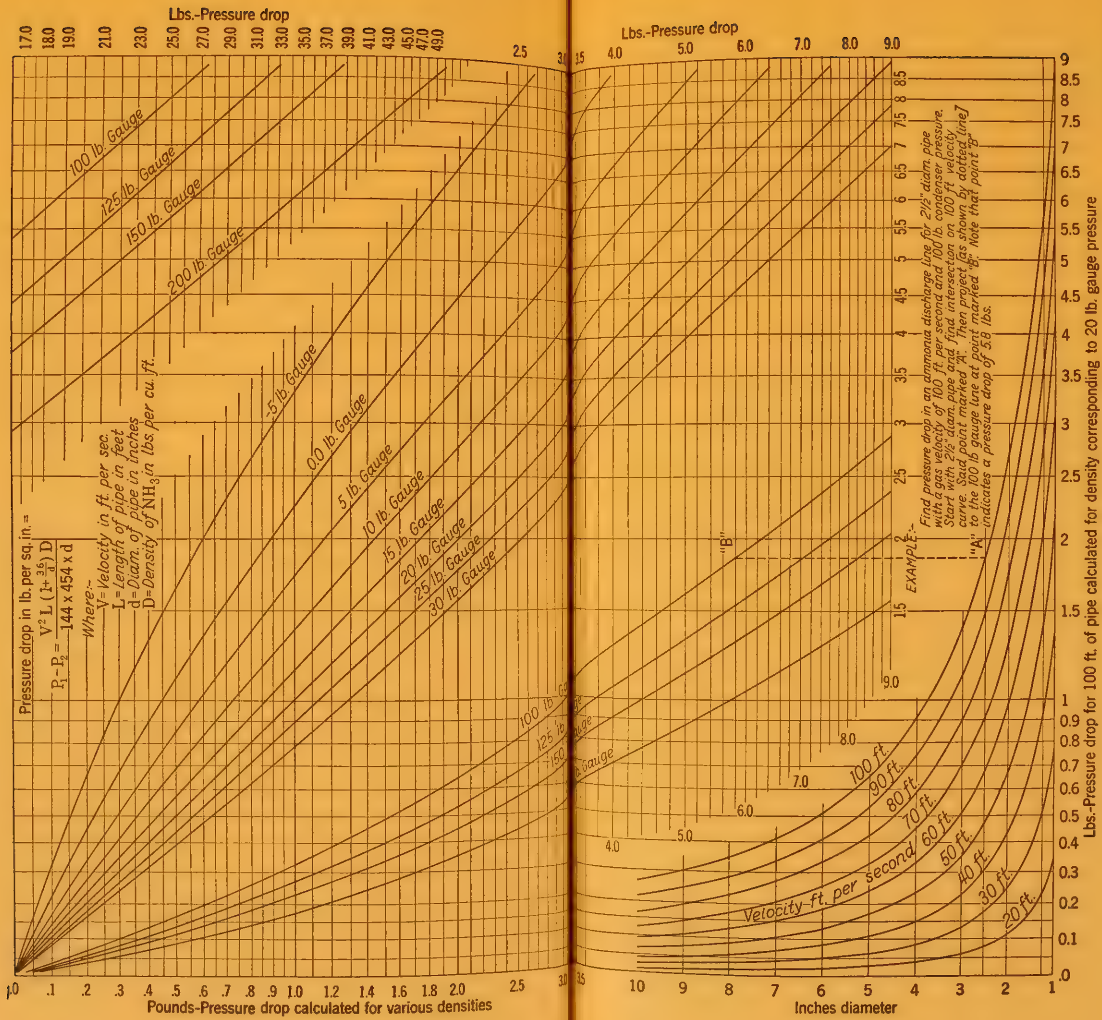

FIG. 227.—Loss of Pressure Due to Flow of Ammonia in Pipe Lines.

of direction are necessary they should be accomplished by means of long radius bends.

Size of the Discharge Pipe.—Refrigerating engineers have not paid much attention to the discharge ports, passages, and the diameter of the piping, but if the compressor horse power is to be reduced to a minimum value the subject of the pressure drop from the cylinder to the condenser must be gone into carefully. It has been the custom to calculate the *average* gas velocity in the discharge from the expression:

$$\text{Average gas velocity} = \frac{\text{piston area} \times \text{stroke} \times 2 \times \text{r.p.m.}}{\text{cross-sectional area of the pipe}}$$

$$= 6000 \text{ to } 7000 \text{ ft. per min.,}$$

if the compressor is double-acting or if it is a twin cylinder, single-acting machine. This calculated value is really only an average gas velocity, whereas the actual velocity is a maximum near mid-stroke and is equal to zero at the dead centers. The result of the intermittent discharge is that the gas pressure is built up in the discharge line and to a lesser extent in the condenser as these volumes act in a manner similar to the air chamber on the discharge line of pumps and compressors. The excess pressure, over and above that of the condenser pressure, shown on many indicator diagrams, is due to the friction to flow and the momentary discharge into a chamber of constant volume.

Size of the Suction Line.—The size of the suction pipe line is very much more important than that of the discharge line. If undue losses of pressure occur in the discharge line the effect will be a decreased volumetric efficiency of the compressor as well as an increased work of compression. However, wire-drawing in the suction line drops the capacity of the compressor rapidly (Fig. 29), increases the horse power per ton of refrigeration and decreases the volumetric efficiency. The results are that, if necessary, the discharge line may be designed for high average gas velocities, but it does not pay to permit velocities greater than 5000 ft. per min. in the suction, and long lines may be as low as 2000 ft. per min., with an average of 3000 ft. An idea can be secured of the loss of pressure from Fig. 227 and the following problem:

Problem.—A 2-in. pipe line for a suction pressure of 20 lb. gage is 300 ft. long. Find the pressure drop if the gas velocity is 5000 ft. per min. Referring to the figure it will be seen that the drop per 100 ft. is 1.85 lb., and the total drop will be 3×1.85 or 5.55 lb.

Condenser Piping.—The details of condenser piping depend on the kind of condenser, some of which are shown in Figs. 228 to 230. When

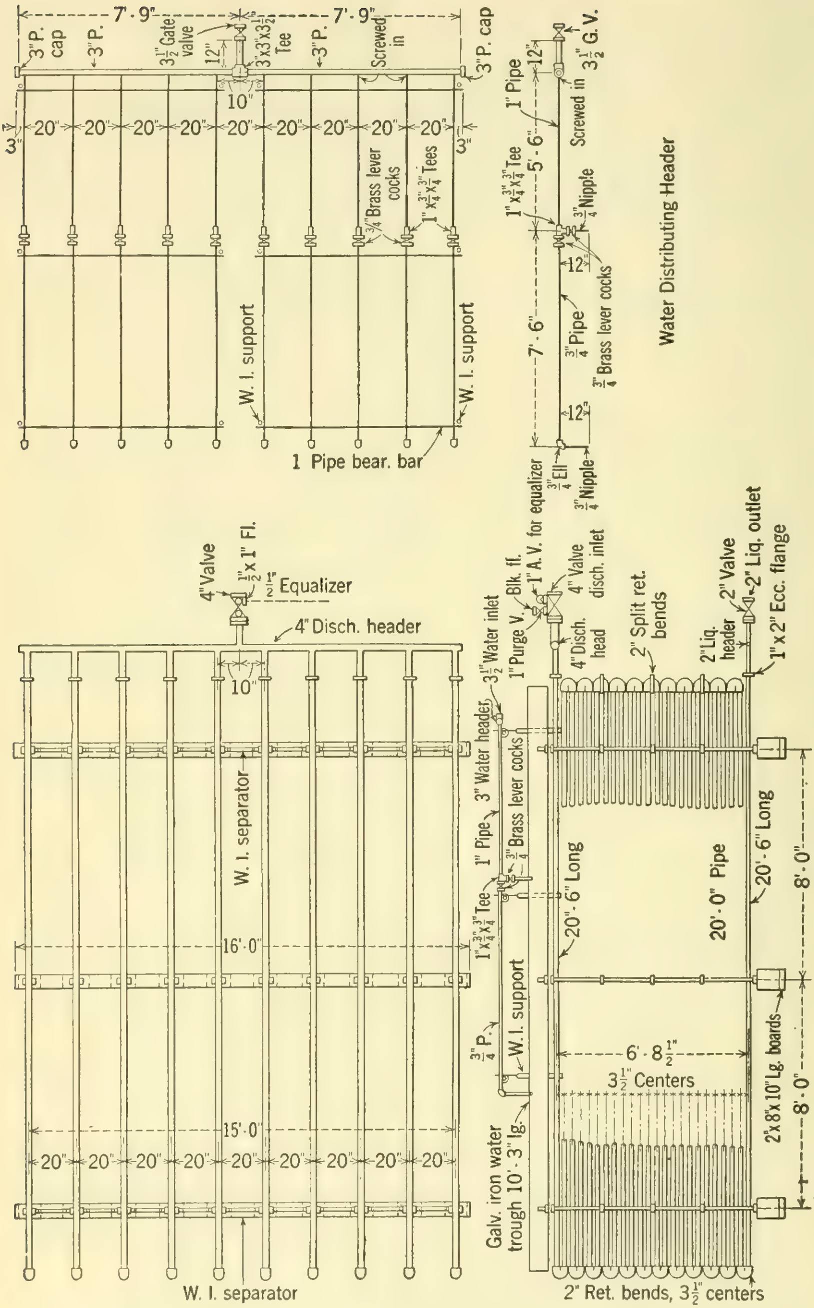

FIG. 228.—Condenser Piping—The Atmospheric Condenser.

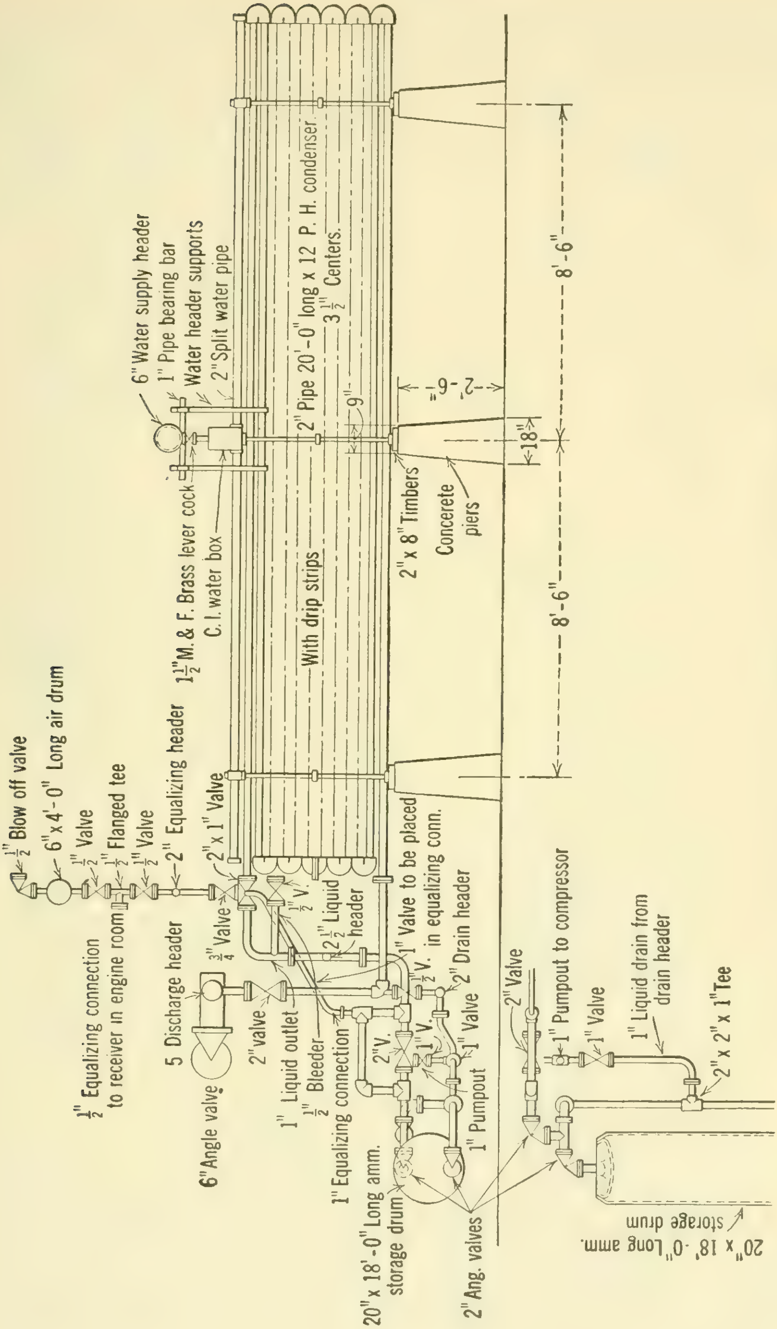

FIG. 229.—Condenser Piping—The Bleeder Condenser.

just one stand, or its equivalent, is used the piping is the simplest, but otherwise the discharge gas from the compressor enters a distributing header and the condensate flows through a collecting header to the liquid

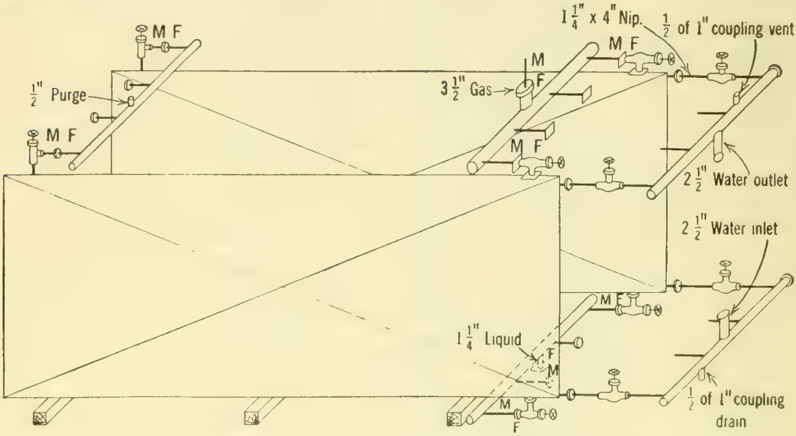

FIG. 230.—Condenser Piping—The Double Pipe Condenser.

receiver. Some manufacturers have a single stop valve on each header, as shown in Fig. 228, while others have stop valves on each stand to facilitate repairs on the condenser, as suggested in Fig. 230. The

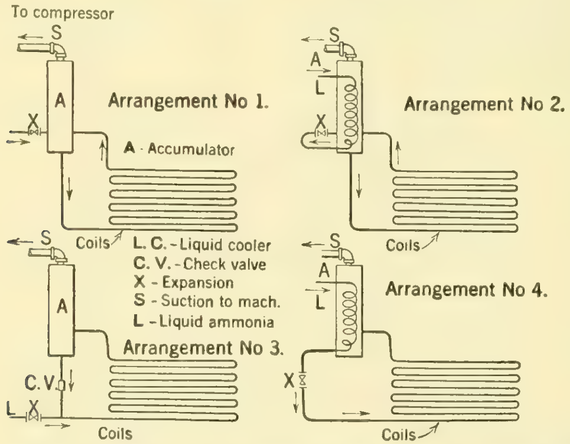

FIG. 231.—Four Piping Arrangements for the Flooded System.

condenser should be provided with connections for equalizing the pressure, with a purge valve, and especially in the large condensers, with a pump-out connection to the main compressor or to a special pump-out

compressor. The water supply connections are indicated in the figures also.

The Liquid Line.—The advisability of having a low temperature liquid refrigerant is shown elsewhere. Some plants are equipped with special liquid coolers, usually a double pipe arrangement for counterflow and with the coldest water in the plant, but this is not common.

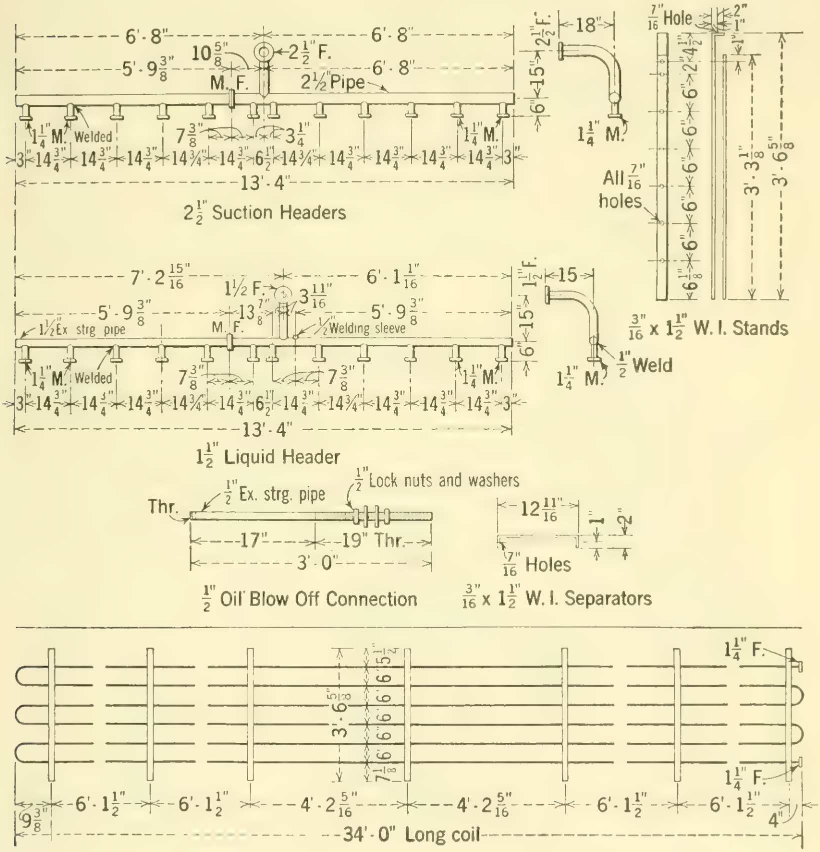

FIG. 232.—Piping and Headers for Can Ice Manufacture.

Neither is it common practice to insulate the liquid line, though this line is sometimes long and frequently placed in hot pipe shafts. There is no question but that insulation is advisable in many cases.

The size of the liquid pipe also depends on the length of the run. As a rule the velocity of flow of 3 to 6 ft. per sec. should be used. Short, self-contained plants can increase this value, and long lines, especially

where considerable vertical rise is required, should be designed using a minimum value. Too large a vertical rise should not be attempted, though on account of the lesser density of the liquid ammonia it will rise 50 per cent higher than will water with the same pressure. The loss of pressure at the expansion valve will be occasioned by pipe friction, velocity head and vertical lift.

The Low Pressure Side.—*Brine cooling.*—Whenever brine is cooled by the use of the shell and tube brine cooler for indirect refrigeration, the simplest pipe arrangement for the ammonia is provided (see Chapter VIII). It is self-contained and there is little danger of liquid return in

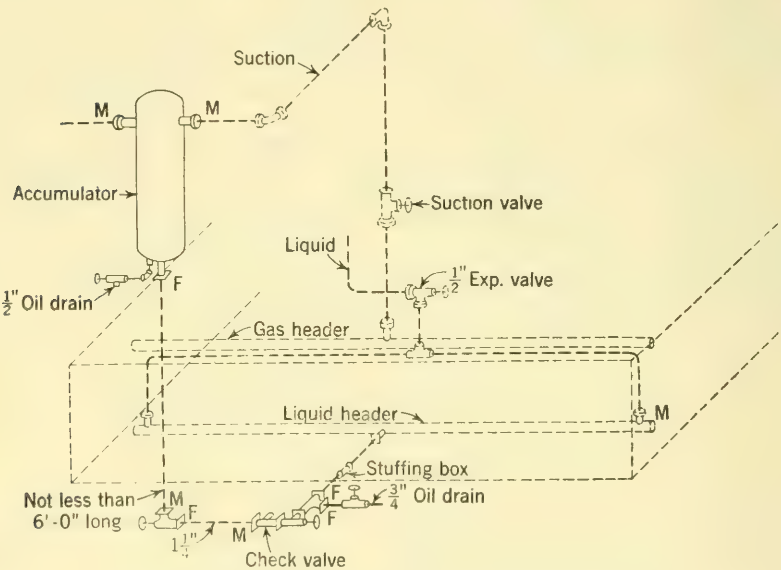

FIG. 233.—The Accumulator and Brine Tank Piping.

the suction pipe line to the compressor. In can ice making it is somewhat more complicated because of the use of the flooded system when direct expansion piping is used for the cooling of brine. Several piping arrangements for the flooded system are shown in Fig. 231. The flooded system embodies a device for precooling the liquid before it enters the expansion coils, and an accumulator to prevent liquid returning to the compressor thereby acting somewhat like the steam separator in steam engineering.

The brine itself is cooled by expansion piping, as shown by Fig. 232. In this case what is desired is a uniform temperature of the brine with a minimum obstruction to the cans. Because of the difficulty of getting

at the piping during operation it is well to use a welded pipe wherever practicable. Limits to welding lie in the ability to handle the coils unless the piping can be welded in place on the job. In can ice making plants using shell and tube brine coolers, the piping would be the same as in the indirect brine system.

Cold Storage Piping.—In cold storage piping the important factors are to secure the proper amount of piping with the least obstruction to

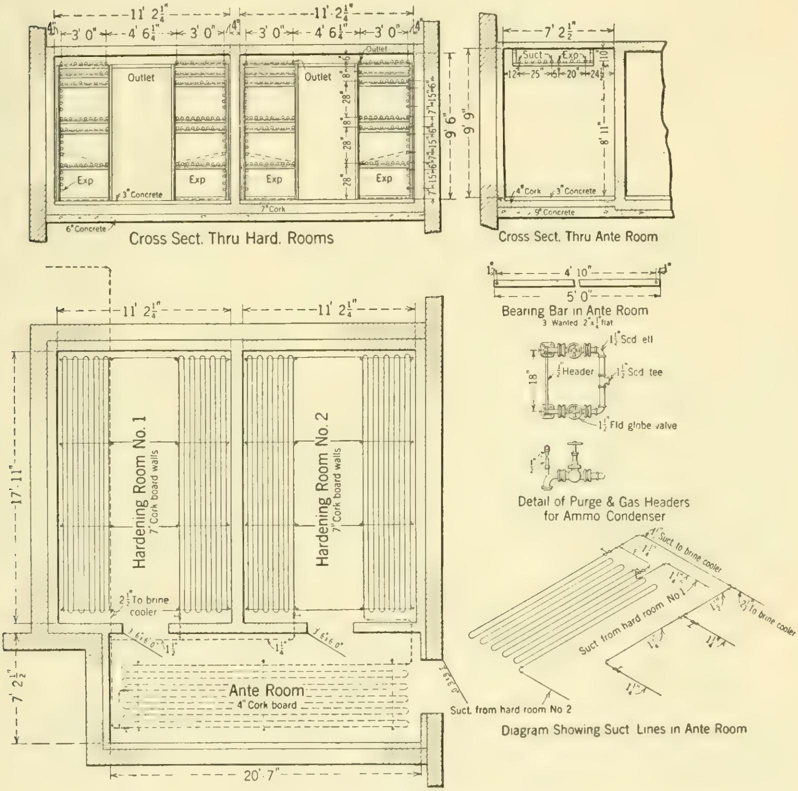

FIG. 234.—Piping—The Sharp Freezer.

the storage of commodities and the least danger due to the melting of the frostation. Typical examples are shown in Figs. 234, 235 and 236 for small rooms and in Figs. 237 to 239 for larger rooms. It will be noticed that the larger rooms use overhead coils entirely. If the piping is not very heavy (in surface) it is usually arranged to be over aisles, but in any case galvanized iron drip pans should be placed under the coils with a suitable lead-off pipe to take the drips from the pipes should the

frostation melt off. How much pipe in series is an important matter to decide. The deciding factor is the ability of the gas to free itself and

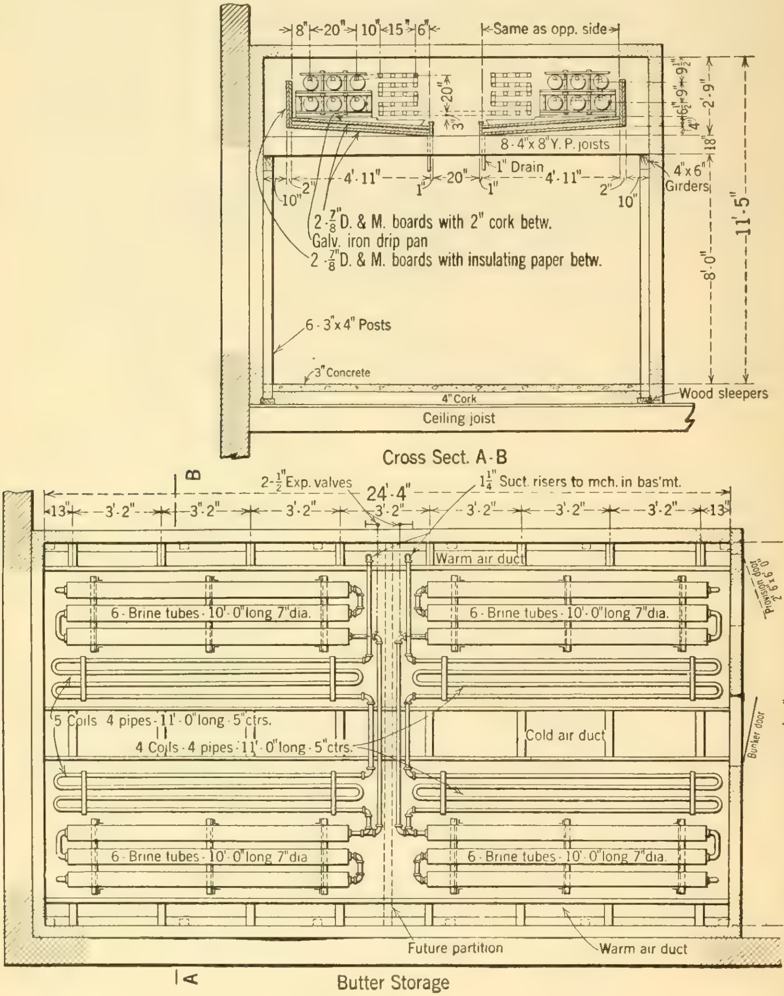

Fig. 235.—Piping—Butter Storage Using Brine Tubes.

return to the compressor without excessive pressure loss. The amount recommended is, as a maximum,

1100 ft. of 1-in. pipe,
1300 ft. of $1\frac{1}{4}$-in. pipe,
1600 ft. of $1\frac{1}{2}$-in. pipe,
1900 ft. of 2-in. pipe.

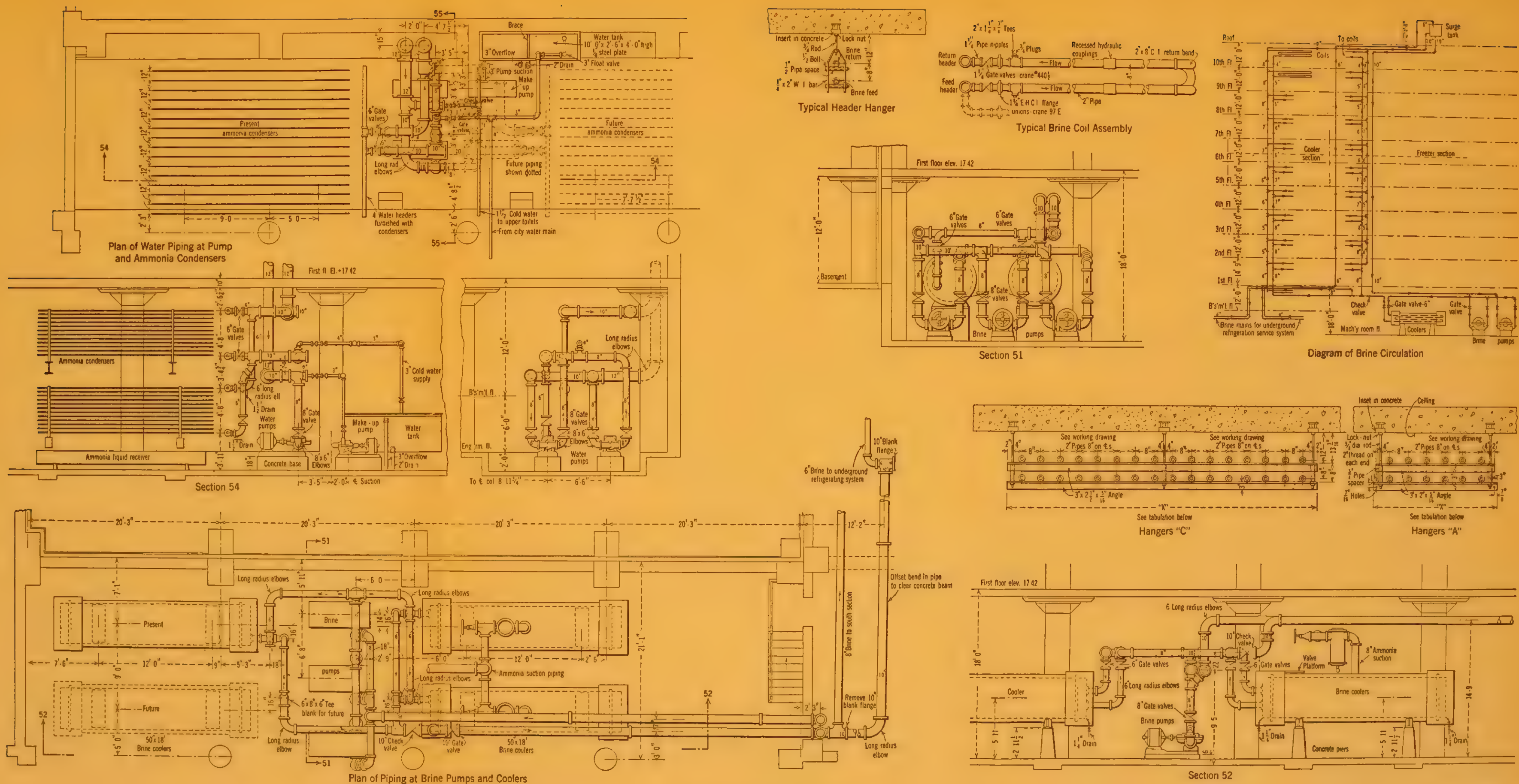

Fig. 237.—Piping—Brine Details.

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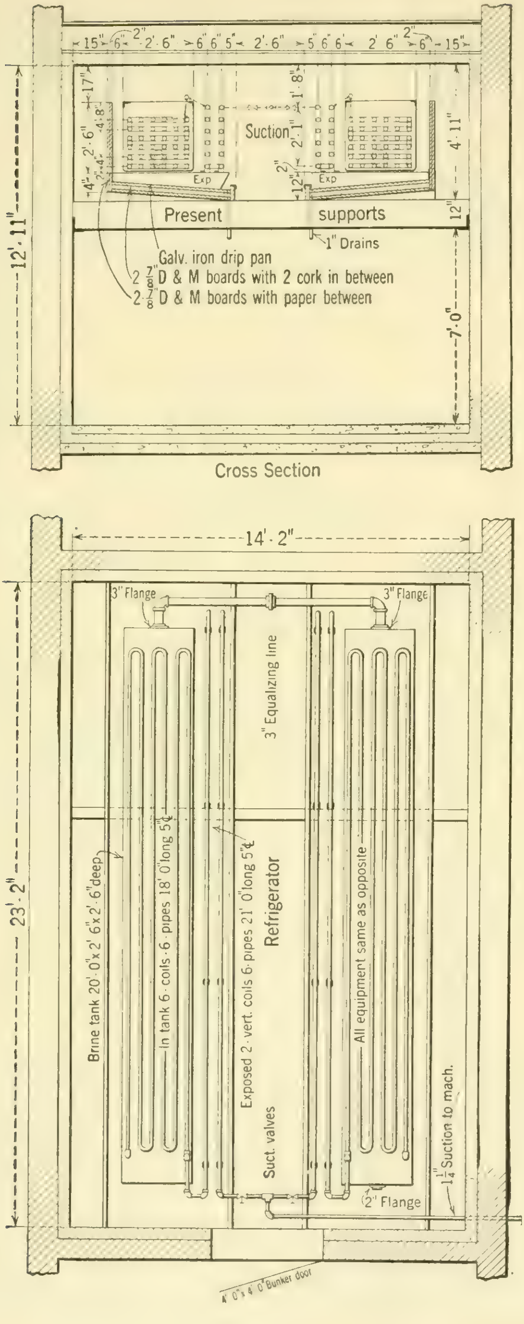

FIG. 236.—Piping—Direct Expansion and Brine Tank.

On the other hand, the pipe line must not be too short. The liquid fed into the pipe line must have an opportunity to boil out before it has to leave the evaporating coils. If it is not all evaporated some liquid will return to the compressor, an action that always causes operating trouble. With brine lines the case is different, as the brine is not subject to the bad effects from gas formation in the pipe. The length of pipe in series for brine varies with the operating conditions from 100 to 400 ft. The use of " top feed " and " bottom feed " for ammonia has been a disputed point for years. At present it is conceded by most engineers, if

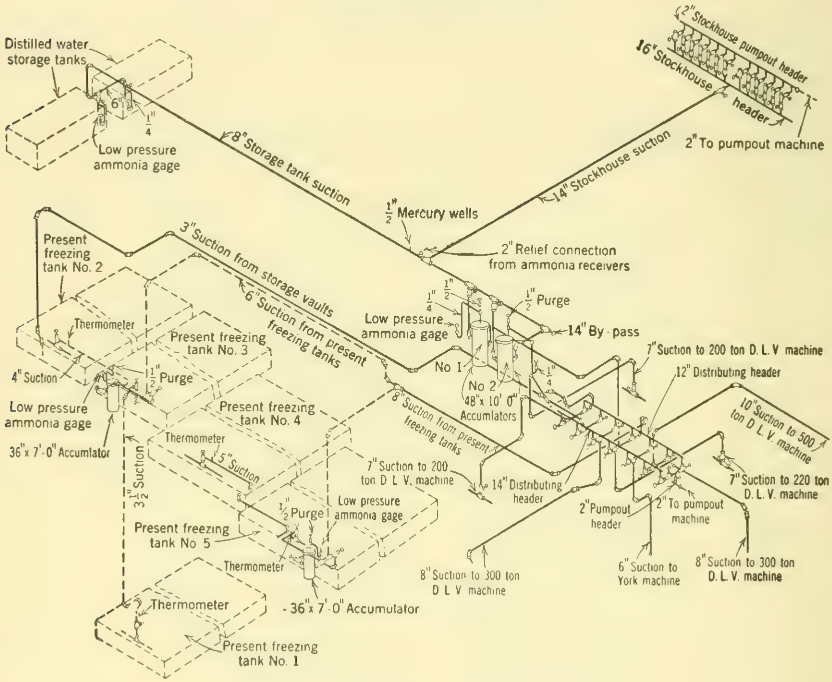

FIG. 239.—Piping—Ammonia Mains.

a suitable protection against liquid returning to the compressor is installed, that bottom feed gives better heat transfer and is the better method of operation. The arrangement of the piping should be such as would permit control of the temperature by increasing or decreasing the amount of piping to suit. This can be done by the arrangement of the coils so that more or less can be turned into action and enough must be supplied to carry the full load.

The pipes are supported in a number of ways. For cold storage work the usual manner is to suspend them from the ceiling. In the case of

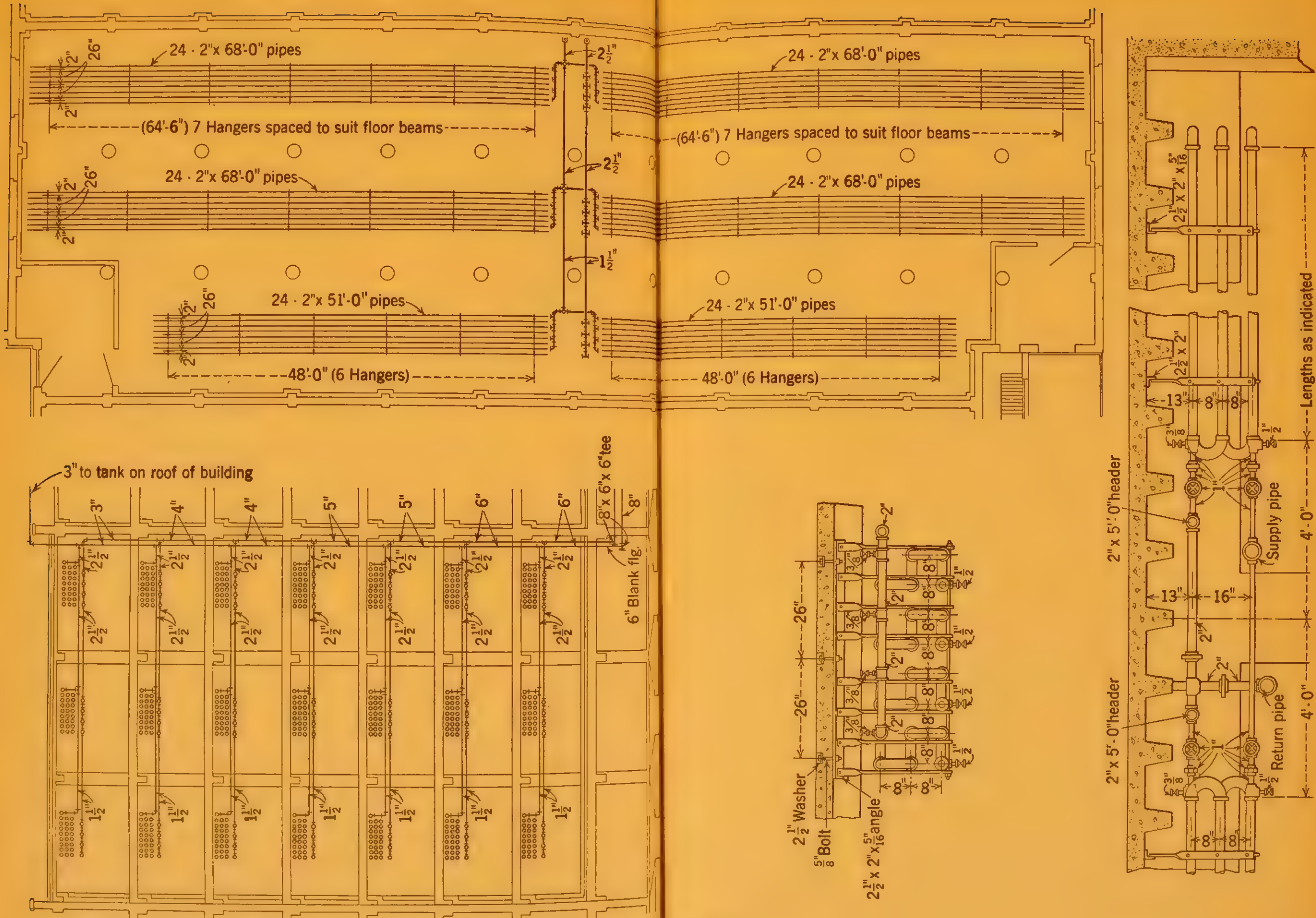

FIG. 238.—P[illegible] Brine Details.

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pipe decks, suitable stands from the decks can be used, and in some special cases the piping may be supported from the walls and from the floor. Examples of piping supports are given in the figures.

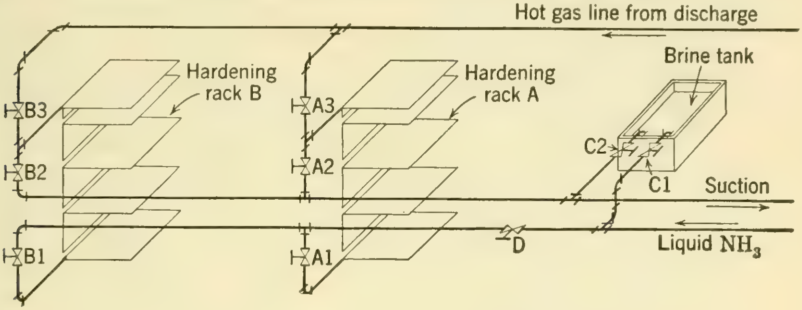


FIG. 240.—Piping—Ice Cream Hardening.

The *ice cream hardening* room is a problem by itself. As usually designed it is very heavily piped (Fig. 240 and Chapter XIV), and the piping is arranged so as to make shelves on which the ice cream is placed in bricks, fancy shapes and in bulk. It is very nearly always a direct

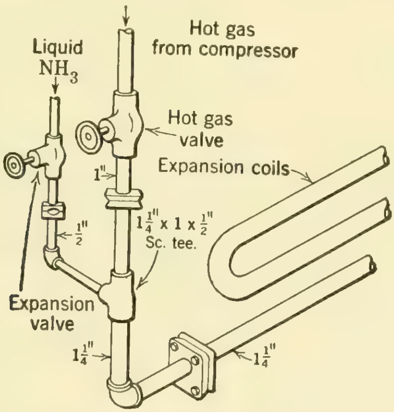


FIG. 241.—Piping—Defrosting Large Installations.

expansion job with the expansion valve arranged for bottom feed of the liquid. The sharp freezer is a term used for a low temperature room. It also is very heavily piped,[1] and may or may not have the piping arranged in the form of shelves. A fish freezer is likely to have shelves so that the fish may be placed in pans on the piping.

Defrosting.—Defrosting of the evaporating coils can be accomplished in a number of ways. The frost may be scraped off from time

[1] See Chapter XIV for the usual values for the amount of piping to be used.

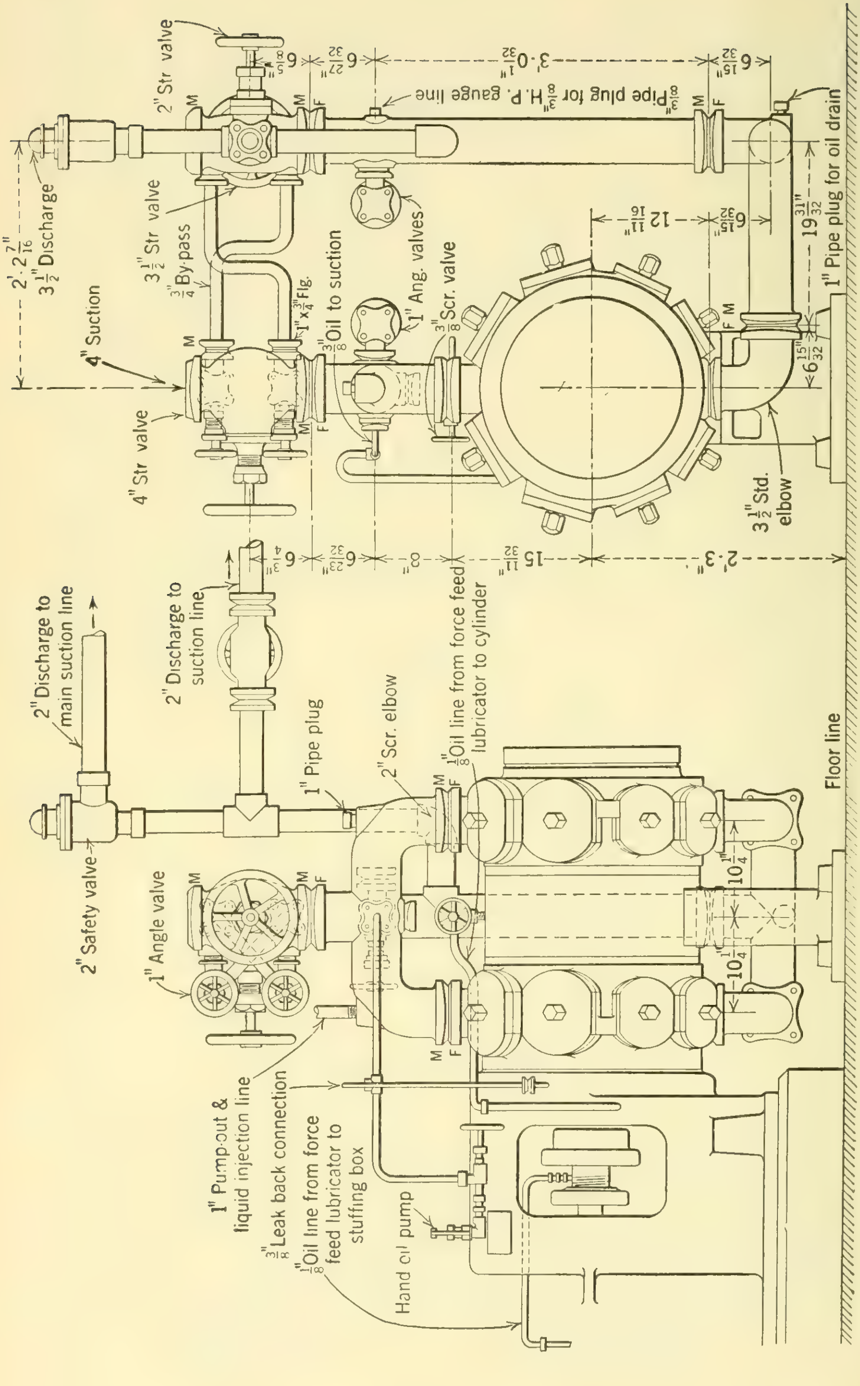


FIG. 242.—Piping—The Horizontal Double Acting Compressor.

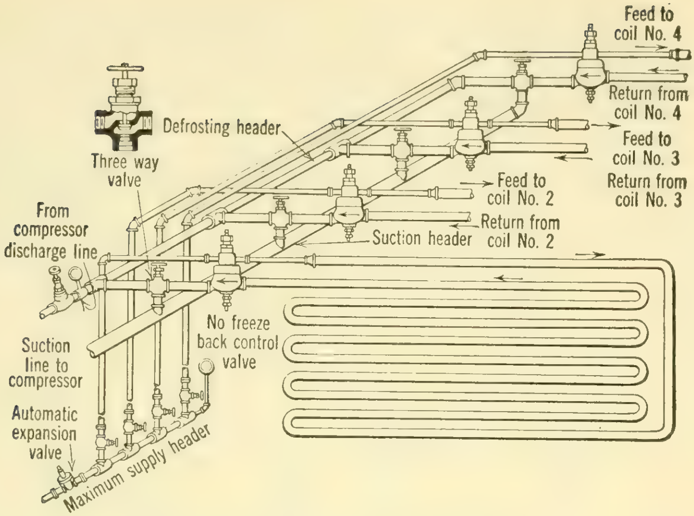


FIG. 243.—Piping—The Non-Freeze Back.

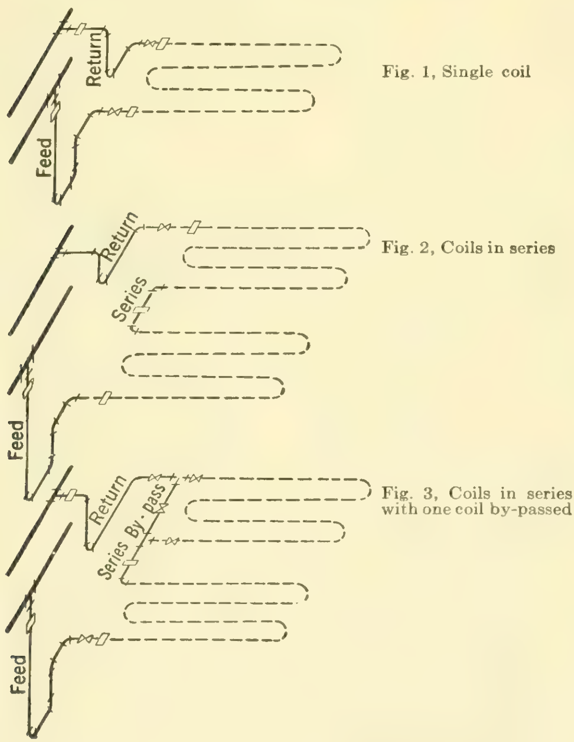


FIG. 244.—Piping Connections to Direct Expansion Coils.

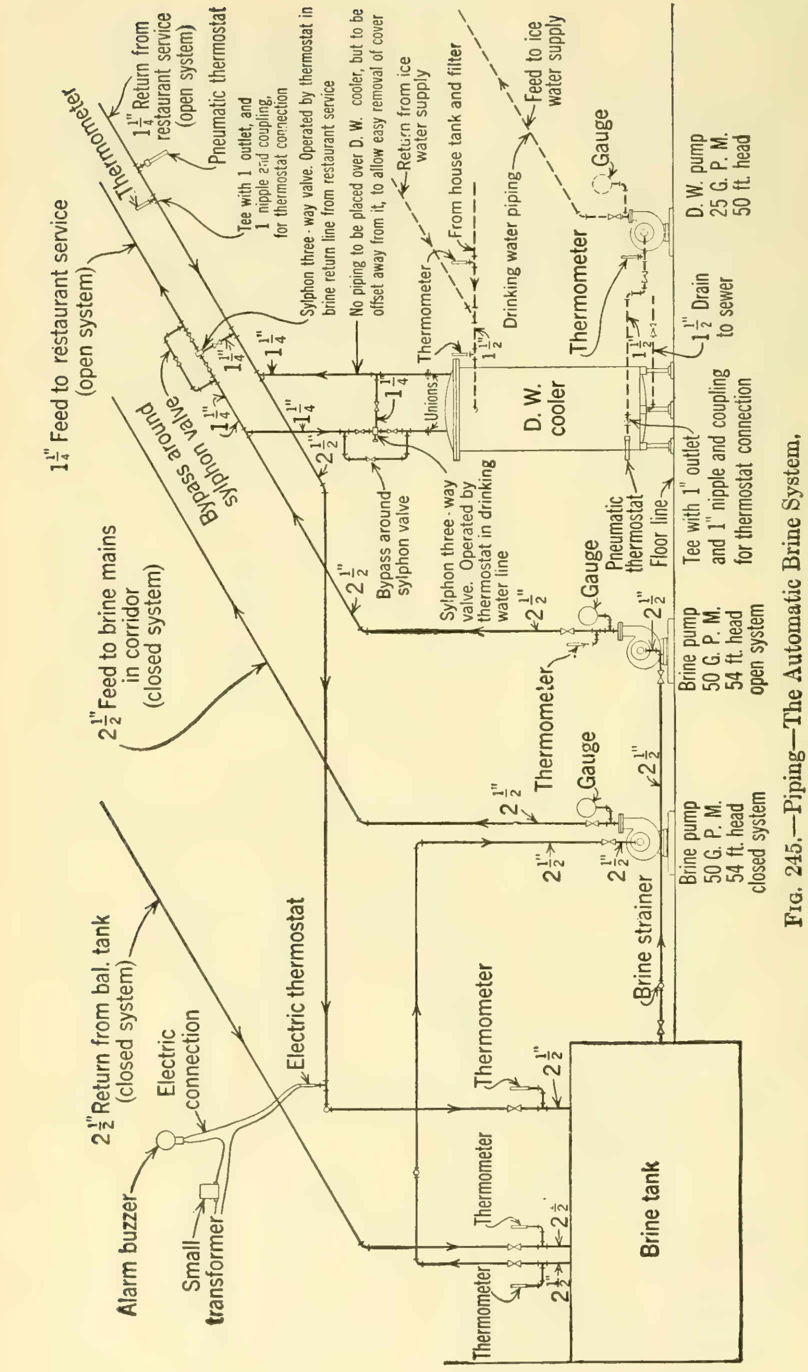


FIG. 245.—Piping—The Automatic Brine System.

to time, and occasionally brine may be arranged to trickle over the pipes. By reversing the connections on the compressor, hot gas can be pumped into the low pressure coils (Fig. 234) or a separate defrosting, and pump-out line can be installed.

Piping for defrosting is shown in Figs. 240, 241 and 243. These are arrangements for permitting hot gas from the compressor to enter the frosted piping.

Non-freeze Back.—A control to prevent automatically the return

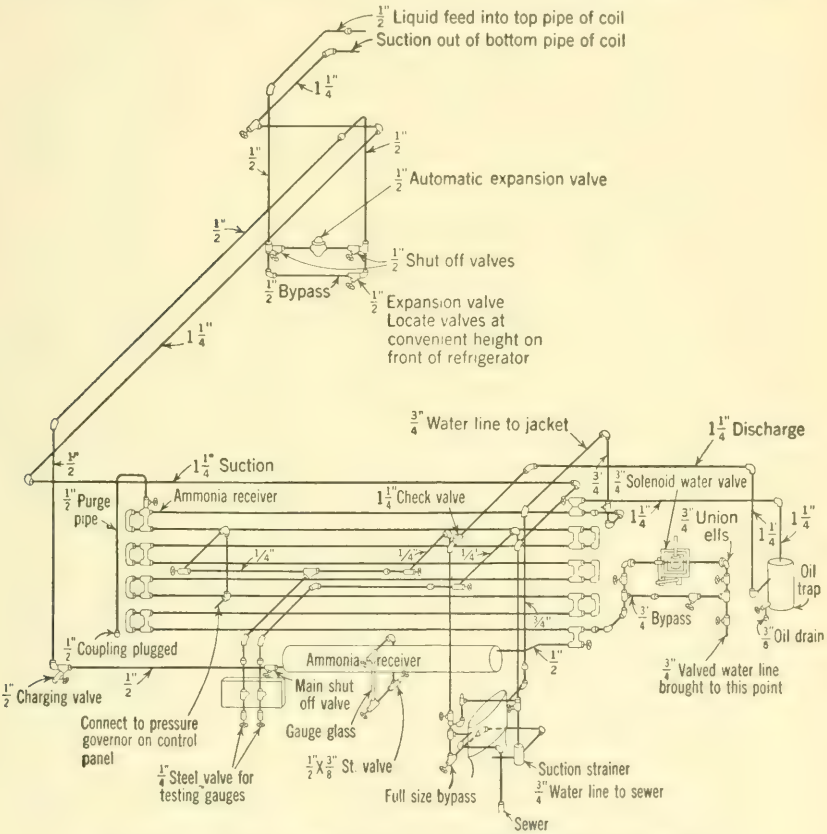


FIG. 246.—Piping—The Automatic Direct Expansion System.

of liquid to the compressor can be accomplished by a control of the temperature in the suction return line. This is done by the device shown in Fig. 236, or the same result can be obtained by means of an automatic control, by winding the bulb of the thermostat around the suction pipe and by setting the contacts for the temperature required in conjunction with a device for controlling the liquid feed.

TABLE 85

Dimensions of Standard Steel Pipe

Diameter, Inches			Thickness, Inches	Circumference, Inches		Transverse Areas, Square Inches			Length of Pipe per Square Foot		Length of Pipe Containing One Cubic Foot, Feet	Weight per Foot of Length, Pounds	Number of Threads per Inch of Screw	Contents per Foot of Length, Gallons
Nominal internal	Actual external	Actual internal		External	Internal	External	Internal	Metal	External surface, feet	Internal surface, feet				
1/8	.405	.269	.068	1.272	.848	.129	.0573	.0717	9.44	14.15	2513.0	.244	27	.0006
1/4	.54	.364	.088	1.696	1.144	.229	.1041	.1249	7.075	10.49	1383.3	.424	18	.0026
3/8	.675	.493	.091	2.121	1.552	.358	.1917	.1663	5.657	7.73	751.2	.567	18	.0057
1/2	.84	.622	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.850	14	.0102
3/4	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3.637	4.635	270.0	1.130	14	.0230
1	1.315	1.049	.133	4.131	3.292	1.358	.8626	.4954	2.904	3.645	166.9	1.678	11½	.0408
1¼	1.66	1.38	.14	5.215	4.335	2.164	1.496	.668	2.301	2.768	96.25	2.272	11½	.0638
1½	1.9	1.16	.145	5.969	5.061	2.835	2.038	.797	2.01	2.371	70.66	2.717	11½	.0918
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42.91	3.652	11½	.1632
2½	2.875	2.469	.203	9.032	7.753	6.492	4.784	1.708	1.328	1.547	30.1	5.793	8	.2550
3	3.5	3.068	.216	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19.5	7.575	8	.3673
3½	4.0	3.548	.226	12.556	11.146	12.566	9.887	2.679	.955	1.077	14.57	9.109	8	.4998
4	4.5	4.026	.237	14.137	12.648	15.904	12.73	3.174	.849	.949	11.31	10.790	8	.6528
4½	5.0	4.506	.247	15.708	14.162	19.635	15.961	3.674	.764	.848	9.02	12.538	8	.8263
5	5.563	5.047	.258	17.477	15.849	24.306	19.99	4.316	.687	.757	7.2	14.617	8	1.020
6	6.625	6.065	.28	20.813	19.054	34.472	28.888	5.584	.577	.63	4.98	18.974	8	1.469
7	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.501	.544	3.72	28.654	8	1.999
8	8.625	7.981	.322	27.096	25.076	58.426	50.04	8.386	.443	.478	2.88	28.554	8	2.611
9	9.625	8.941	.342	30.238	28.076	72.76	62.73	10.03	.397	.427	2.29	33.907	8	3.300
10	10.75	10.020	.365	33.772	31.477	90.763	78.839	11.924	.355	.382	1.82	40.483	8	4.081
11	11.75	11.0	.375	37.699	35.343	113.098	99.402	13.696	.318	.339	1.456	45.557	8	
12	12.75	12.0	.375	40.055	37.7	127.677	113.098	14.579	.299	.319	1.27	49.562	8	

13	14.0	13.25	.375	43.982	41.626	153.938	137.88	16.051	.273	.288	1.04	54.568	8	
14	15.0	14.25	.375	47.124	44.768	176.715	159.485	17.23	.255	.268	.903	58.573	8	
15	16.0	15.25	.375	50.265	47.909	201.062	182.655	18.407	.239	.250	.788	62.579	8	
17	18.0	17.25	.375	56.549	54.192	254.47	233.706	20.764	.212	.221	.616	70.589		
19	20.0	19.25	.375	62.832	60.476	314.16	291.04	23.12	.191	.198	.495	78.599		
21	22.0	21.25	.375	69.115	66.759	380.134	354.657	25.477	.174	.179	.406	86.609		
23	24.0	23.25	.375	75.398	73.042	452.39	424.558	27.832	.159	.164	.339	94.619		

Dimensions of Extra-heavy Steel Pipe

Diameter, Inches			Thickness, Inches	Nearest Wire Gage, Number	Circumference, Inches		Transverse Areas, Square Inches			Length of Pipe, per Square Foot		Nominal Weight per Foot, Pounds
Nominal internal	Actual external	Actual internal			External	Internal	External	Internal	Metal	External surface, feet	Internal surface, feet	
1/8	.405	.215	.095	12½	1.272	.644	.129	.033	.086	9.433	18.632	.314
1/4	.54	.302	.119	11	1.696	.924	.229	.068	.161	7.075	12.986	.535
3/8	.675	.423	.126	10½	2.121	1.323	.358	.139	.219	5.657	9.07	.738
1/2	.84	.546	.147	9	2.639	1.703	.554	.231	.323	4.547	7.046	1.087
3/4	1.05	.742	.154	8½	3.299	2.312	.866	.4324	.414	3.637	5.109	1.473
1	1.315	.957	.179	7	4.131	2.988	1.358	.71	.648	2.904	4.016	2.171
1¼	1.66	1.278	.191	6½	5.215	3.996	2.164	1.271	.893	2.301	3.003	2.996
1½	1.9	1.500	.200	6	5.969	4.694	2.835	1.753	1.082	2.01	2.556	3.631
2	2.375	1.939	.218	5	7.461	6.073	4.43	2.935	1.495	1.608	1.975	5.022
2½	2.875	2.323	.276	2	9.032	7.273	6.492	4.209	2.283	1.328	1.649	7.661
3	3.5	2.900	.300	1	10.996	9.085	9.621	6.569	3.052	1.091	1.328	10.252
3½	4.0	3.364	.318	0	12.566	10.549	12.566	8.856	3.71	.955	1.137	12.505
4	4.5	3.826	.337	0	14.137	11.995	15.904	11.449	4.455	.849	1.0	14.983
5	5.563	4.813	.375	00	17.477	15.120	24.306	18.193	6.12	.687	.793	20.778
6	6.625	5.761	.432	000	20.813	18.064	34.472	25.967	8.505	.577	.664	28.573

The Radiator Section.—The radiator section, made of a thin walled cast iron or semi-steel, can be manufactured to hold ammonia, and it has become popular in certain sections of the United States for certain kinds of refrigeration. For ammonia the section is made with the tongue-and-groove flanged joint, and can be erected in the usual manner (as for heating installations) with the columns vertical, or, for shelving in ice cream hardening rooms, with the columns horizontal. Tests[2] indicate that a section of this nature will withstand a heavy hydraulic

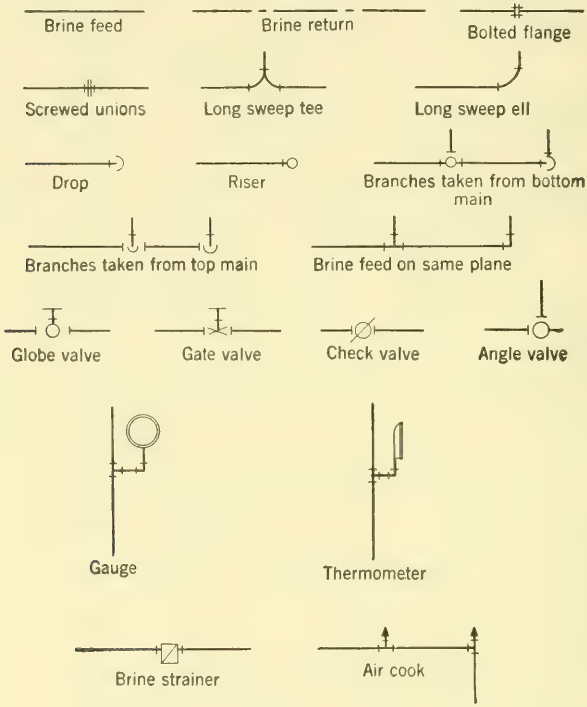


FIG. 247.—Conventional Piping Symbols.

pressure and will carry a heavy load on a long span without leaks or without showing signs of failure. The advantages of cast sections are in the ease of erection, the probable better heat transfer[3] and the greater ease with which the section frees itself of gas as compared with common piping. The application of cast iron or semi-steel sections to ammonia refrigeration is similar to the arrangement used in steam heating.

[2] The Flexibility of Cast Iron Sections for Ammonia, H. J. Macintire, Journal of the American Society of Refrigerating Engineers, 1922.

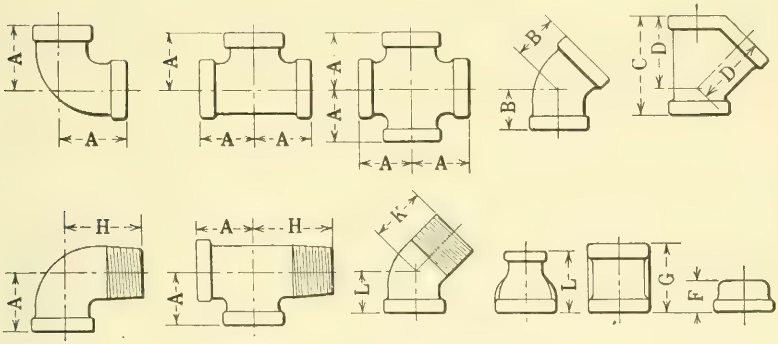
[3] Heat Transfer in Cast Iron Sections, H. J. Macintire, Refrigerating Engineering, Dec., 1924.

Figures 245 and 246 show typical piping for automatic operation of brine and ammonia systems. Table 85 gives the dimensions and cross-sectional areas of full weight and extra heavy wrought iron and steel pipe, and Table 86 gives the general dimensions of standard malleable iron screw fittings. Figures 247 gives typical conventional symbols for piping.

TABLE 86

Standard Malleable Iron-screwed Fittings

General Dimensions



All Dimensions Given in Inches

Size	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$	5	6	7	8
A	$\frac{11}{16}$	$\frac{13}{16}$	$\frac{15}{16}$	$1\frac{1}{8}$	$1\frac{5}{16}$	$1\frac{7}{16}$	$1\frac{3}{4}$	$1\frac{15}{16}$	$2\frac{1}{4}$	$2\frac{11}{16}$	$3\frac{1}{8}$	$3\frac{7}{16}$	$3\frac{3}{4}$	$4\frac{1}{16}$	$4\frac{7}{16}$	$5\frac{1}{8}$	$5\frac{13}{16}$	$6\frac{1}{2}$
B	...	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{5}{16}$	$1\frac{7}{16}$	$1\frac{11}{16}$	$1\frac{15}{16}$	$2\frac{3}{16}$	$2\frac{3}{8}$	$2\frac{5}{8}$	$2\frac{13}{16}$	$3\frac{1}{16}$	$3\frac{7}{16}$	$3\frac{7}{8}$	$4\frac{1}{4}$
C	...	...	...	$2\frac{1}{2}$	$2\frac{7}{8}$	$3\frac{7}{16}$	$4\frac{1}{16}$	$4\frac{1}{2}$	$5\frac{7}{16}$	$6\frac{1}{4}$	$7\frac{1}{4}$	...	$8\frac{7}{8}$					
D	...	...	...	$1\frac{11}{16}$	2	$2\frac{7}{16}$	$2\frac{15}{16}$	$3\frac{5}{16}$	$4\frac{1}{16}$	$4\frac{11}{16}$	$5\frac{9}{16}$	...	$6\frac{15}{16}$					
E	...	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{7}{16}$	$1\frac{11}{16}$	$2\frac{1}{16}$	$2\frac{5}{16}$	$2\frac{13}{16}$	$3\frac{1}{4}$	$3\frac{11}{16}$	4	$4\frac{3}{8}$					
F	...	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{7}{16}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{15}{16}$	2	...	$2\frac{5}{16}$	$2\frac{9}{16}$		
G	...	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{1}{2}$	$1\frac{11}{16}$	$1\frac{15}{16}$	$2\frac{1}{8}$	$2\frac{1}{2}$	$2\frac{7}{8}$	$3\frac{3}{16}$							
H	$1\frac{1}{8}$	$1\frac{5}{16}$	$1\frac{7}{16}$	$1\frac{5}{8}$	$1\frac{7}{8}$	$2\frac{1}{8}$	$2\frac{1}{2}$	$2\frac{11}{16}$	$3\frac{3}{16}$	$3\frac{13}{16}$	$4\frac{1}{2}$	...	$5\frac{11}{16}$					
K	...	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{1}{2}$	$1\frac{11}{16}$	$1\frac{7}{8}$	$2\frac{1}{4}$	...	3	...	$3\frac{3}{4}$					
L	...	$\frac{11}{16}$	$\frac{11}{16}$	$\frac{13}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{11}{16}$	...	$2\frac{1}{8}$							

Do not use for reducing fittings

CHAPTER XIII

ICE MAKING

Natural Ice.—Natural ice is less used every year. Where a few years ago it was the chief refrigerating agent it has now become almost negligible, as can be seen from the table (Table 87) which gives the amount of natural ice cut and stored for the last twenty years. The reasons for this falling off in the cutting of natural ice are several. The first and most important reason, perhaps, is its unreliability. Ice is a necessity, and at times the natural ice crop fails because of an open winter. Under these circumstances the hold-over from the previous winter hardly supplies the demand. Besides this, the cost of transportation is now much greater than heretofore and ice is a bulky commodity. Thus, although the railroads still transport a heavy tonnage, the average distance is not great. A lesser reason, but of some force even then, for the decline in the use of natural ice, is the fear of a contaminated source for the ice. This is, however, not a matter of serious concern as it has been well determined that freezing frees the ice of impurities, and when it is not used for several months the bacteria are mostly killed before the ice is consumed. At present natural ice is used chiefly for car and boat icing, where the appearance of the ice is not a factor.

Manufactured Ice.—The artificial ice industry, including plants, buildings and delivery equipment, is estimated (1924) to comprise nearly a billion dollars' investment. It is placed ninth in point of industrial investment in the United States, and there is reason to believe that the growth is going to continue, as more organized efforts are made to increase ice sales, especially to the household trade.[1]

In 1922 there were listed (Ice and Refrigeration Blue Book) 5500 plants manufacturing ice, with a daily total capacity of 223,000 tons. In this compilation 55.9 per cent was distilled water ice, 41.4 per cent was raw water can ice, and 2.7 per cent was plate ice, and of the plants engaged in the production of ice primarily 85 per cent had compression

[1] The effect of the household refrigerating machine is beginning (1927) to be felt in the larger cities, and in some localities the electric driven two, one and the fractional tonnage compressors have practically replaced ice in the wholesale trade for soda fountain, groceries and markets.

TABLE 87

RECAPITULATION

	Capacity of Tons	Tons Harvested
Total Hudson River Houses	1,823,321	238,083
Total Lakes and Ponds	194,804	4,548
	2,018,125	242,631

RECORD OF PREVIOUS YEARS

Year, Totals	Capacity in Tons	Tons Harvested
1923–1924	2,018,125	242,631
1922–1923	2,320,064	642,851*
1921–1922	2,508,798	1,029,808
1920–1921	2,698,032	689,993†
1919–1920	2,800,939	1,237,409
1918–1919	2,921,927	690,353
1917–1918	3,066,703	2,537,482
1916–1917	3,043,127	2,001,150
1915–1916	3,201,943	1,520,620
1914–1915	3,158,989	1,398,191
1913–1914	3,226,086	1,938,149
1912–1913	3,490,420	1,202,166
1911–1912	3,397,037	2,853,120
1910–1911	3,558,275	2,262,593
1909–1910	3,644,150	2,206,984
1908–1909	3,898,297	1,861,192
1907–1908	3,933,271	2,539,941
1906–1907	4,137,259	3,582,324
1905–1906	4,117,050	1,672,188
1904–1905	4,154,000	3,572,371
1903–1904	4,361,800	3,661,800
1902–1903	4,628,200	2,595,110
1901–1902	4,833,100	3,943,100
1900–1901	4,791,400	4,606,800

* Inclusive of 308,191 tons carried over from previous year.
† Inclusive of 499,170 tons carried over from previous year.

plants, 12.2 per cent had absorption, and 2.7 per cent had both systems. It may be mentioned that nearly every plant installed of late years has been either electric or internal combustion engine-driven, which means that few distilled water systems now go in, and it is doubtful whether a plate system plant has been installed in the last ten years in the United States.

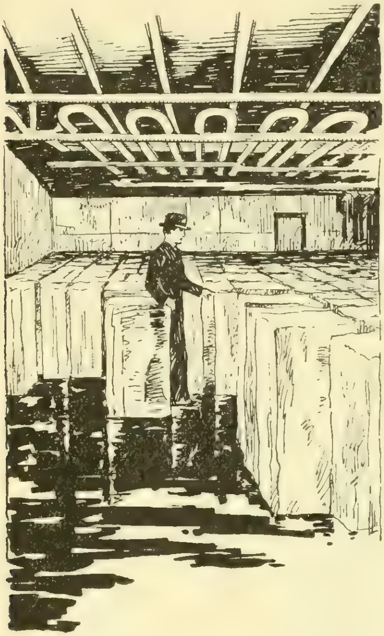


Fig. 248.—Daily Ice Storage.

The ice manufacturing industry, like all applications of refrigeration, has changed from steam engine to oil engine and particularly to electric motor-driven compressors. With the steam engine it was convenient to use the exhaust of the engine as the source of the water to be frozen, under which conditions the system became a distilled water can plant. With the rise in the cost of fuel and labor, and the desirability of the use of electric power—both from its simplicity and also from its cleanliness—there has been a very decided change, and few steam engines are now being installed in new plants except under special circumstances. This means, practically, the impossibility of using the distilled water system and therefore these electric or internal combustion engine-driven compressors are of necessity designed for the raw water can system. The use of electricity also has directly stimulated economies in the cost of ice production, and, in fact, is the greatest single factor in decreasing ice manufacturing costs.

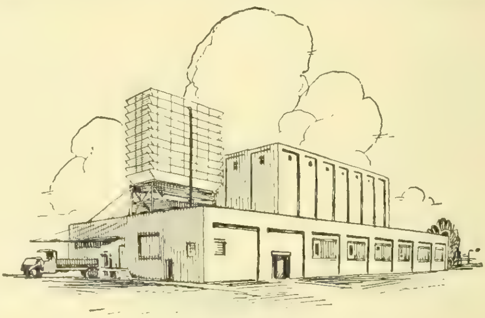


Fig. 249.—Typical Ice Making Plant.

The Distilled Water System.—If well, lake, river or pond water is frozen in cans without the proper system the ice produced will be marble-like in appearance, due to the air which is always in solution in the

water. This marble or opaque ice is good ice as far as any useful purpose is concerned. It melts at 32 deg. F., and its latent heat of fusion is practically the same as crystal ice, but the American public does not like it for table or household purposes. Where crushed ice is used, as in ice cream packing, ice and ice and salt railroad car icing, or in fish packing, there is no difference between the two, but for the retail trade the transparent ice must be supplied.

One of the first successful devices for the manufacture of crystal artificial ice is the distilled water system. The exhaust from the steam engine is condensed (at or near 212 deg. F.) and is brought up to a boil again while exposed to the atmosphere in order to permit the air present in the steam to pass out (Fig. 250.) This so-called reboiler has a skimmer or surface blowoff to remove the surface foreign material, as cylinder oil,

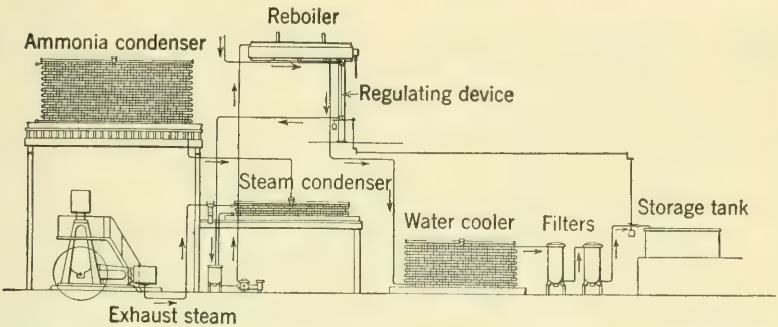


FIG. 250.—The Distilled Water Ice Plant.

etc. The condensate is then cooled by means of water-cooled coils to about 70 to 80 deg. F., and is passed through both a sand and a charcoal filter—the latter to remove any traces of oil present. The distilled water is now ready for the cans unless some system of precooling is available. The resulting ice, if care is taken not to let air become absorbed by the distilled water, will be transparent without the medium of any complicated auxiliary devices. Being free of air and dissolved salts, the water will freeze perfectly clear, unless impurities get into the water and cause a white or a red core. Figure 250 shows the ordinary layout for a distilled water can ice-making system.

There are certain localities where the distilled water plant has to be used. The so-called raw water can ice plant must have good water for the cans. If this water is heavily impregnated with salts of a nature that cannot be precipitated, for example, sodium salts, it is probable that the distilled water plant will have to be used. For such a plant the multiple-effect evaporator may work out satisfactorily (Fig. 251), or it is possible

that the old steam engine-driven compressor arrangement will be the best suited. The problem boils down to the best plan for freeing of the water from the salts in solution, and each case should be solved on its particular merits. While the distilled water system is not being installed in new plants to any great extent at the present time, there is a large number of plants that still use it. However, every year the number of plants changing over to electrically operated systems is large. One disadvantage of the distilled water system is the rapid deterioration of the distilling apparatus aside from the expense of the distillation.

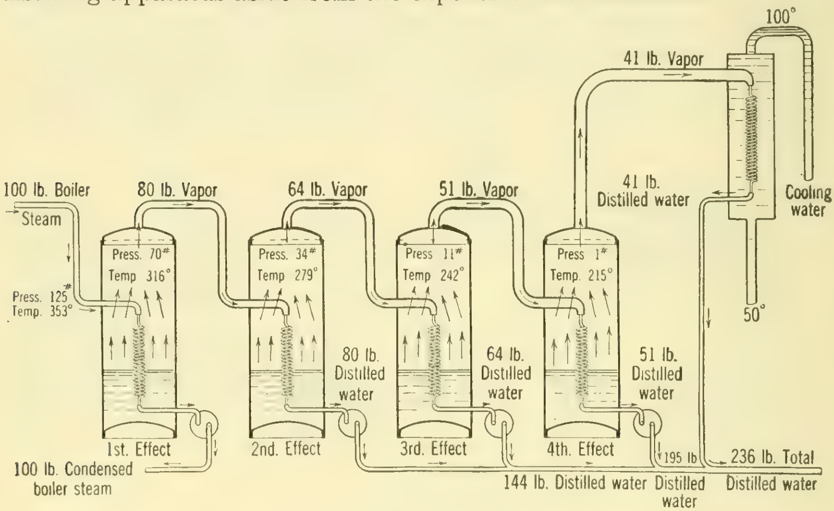


FIG. 251.—Four Effect Distilled Water Evaporator.

The Raw Water System.—One of the first of the so-called raw water systems (water not distilled is called raw water) was the *plate* ice method. This system is one in which the refrigerant maintains a plate at zero deg. F. or lower while it is submerged in a tank of water. Ice is formed on one side only, the water near the freezing surface being in a constant state of agitation. In the process of five or seven days a thickness of some 10 to 11 in. of ice is formed, and this ice is melted off the plate by permitting hot gas from the compressor to pass into the coils or the hollow part of the plate. The plate is quite large and the tank is deep, so that the cake of ice formed, which has to be lifted by a crane, weighs from three to five tons. This large cake is now removed to the saw table where 300-lb. cakes are made up.

The plate[2] system makes good ice, and, in fact, it is said that no

[2] Harry Sloan says (N.A.P.R.E., 1922) that the quality of plate ice has never been equalled by any other method of manufacture. Only slight differences in thick-

better ice can be made. It is non-uniform in thickness, however, and there is some prejudice against it on the part of the retailer. The tank is deeper than the can ice tank would be, and the head room has to be greater. The crane may have to be stronger, although the present tendency in can ice making is to increase the load on the crane, but the saw table involves expense for power and there is the trouble of snow ice formed by the sawing operation. The net result (due to the greater first cost, trade resistance and, as a rule, the greater operating cost) is an absence in the installation of new plate plants during the last few years.

The Can Raw Water System.—The history of raw water can ice making systems shows a large number of different devices. The attempt at first was to prevent opaque ice, due to dissolved air in the water, and later the problem became a desire to overcome the effects of dissolved salts as well as dissolved air in the water. Paddles were used in the early designs, and later a plunger with a pipe connection to the can so that an oscillation of the plunger created a surge of the water in the can. The entire tank, if a small one, has been placed on trunnions and has been rocked by a suitable mechanism. Experiments towards the elimination of all agitation by special treatment has been attempted for years and especially lately.[3] This latest process requires the use of galvanized piping throughout, a telescopic water storage tank, a special can filling device and finally a very careful chemical treatment and filtration followed by a subjection of the water heated to about 150 deg. F. to a very high vacuum. A second filtering completes the process. It is stated that the first and the operating cost is less than is required for high pressure air agitation.

Of all the methods of freezing water for commercial purposes the can system of raw water ice manufacture has proved itself to be the most successful arrangement. The cans are usually 300 or 400 lb., although smaller sizes are used in small tanks or where the ice is desired to be frozen during the day-time operation. The standard ice can size (Table 88) is about 11 in. by 22 in. cross-section and this size in the 300- or 400-lb. cake has been found to be convenient, both as regards freezing time and also as regards the handling and the cutting up for the trade.

In the can system the water to be frozen is put into the cans, and these are lowered into a tank of brine until the brine level (on the outside of the cans) is at least up to the surface of the water inside of the can.

ness and rounded edges appear on the top row of cakes. Much more evenness of thickness has been obtained since the introduction of the flooded system.

[3] Van R. Greene, Eastern Ice Association, April, 1923.

There is an advantage in having the cans submerged below this point, both as regards the shortening of the freezing time required and also as regards the decrease in the probable amount of checking of the ice. The brine for raw water ice is maintained cold at a temperature varying usually from 14 to 18 deg. F. by means of direct expansion piping in the tank or by means of shell and tube cooler submerged in the brine in the tank, and is kept circulated by means of a suitable propeller. The water to be frozen may have excessive salts, and always has a considerable amount of *air*, in solution. The former may be decreased, as regards the weight of actual solids, by the water softening treatment of lime and soda ash or by pumping out the core water after the can is about three-quarters frozen. These solids and the air in solution constitute a real problem, and their removal brings the variety of designs which are now to be found on the market.

TABLE 88

THE STANDARD SIZES OF ICE CANS

Weight of Cake of Ice Pounds	Inside Dimensions, Inches			Length Overall, Inches	Thickness of Material, U. S. Standard Gage	
	Top	Bottom	Length		Sides	Bottom
25	4×9	$3\frac{1}{2} \times 8\frac{1}{2}$	23	24	No. 18	No. 18
50	5×12	$4\frac{1}{2} \times 11\frac{1}{2}$	31	32	No. 16	No. 16
50	6×10	$5\frac{1}{2} \times 9\frac{1}{2}$	31	32	No. 16	No. 16
50	8×8	$7\frac{1}{2} \times 7\frac{1}{2}$	31	32	No. 16	No. 16
60 (25 Kilos.)	5×14	$4\frac{1}{2} \times 13\frac{1}{2}$	31	32	No. 16	No. 16
100	8×16	$7\frac{1}{4} \times 15\frac{1}{4}$	31	32	No. 16	No. 16
200	$11\frac{1}{2} \times 22\frac{1}{2}$	$10\frac{1}{2} \times 21\frac{1}{2}$	31	32	No. 16	No. 16
200	$14\frac{1}{2} \times 14\frac{1}{2}$	$13\frac{1}{2} \times 13\frac{1}{2}$	35	36	No. 16	No. 16
300	$11\frac{1}{2} \times 22\frac{1}{2}$	$10\frac{1}{2} \times 21\frac{1}{2}$	44	45	No. 16	No. 16
400	$11\frac{1}{2} \times 22\frac{1}{2}$	$10\frac{1}{2} \times 21\frac{1}{2}$	57	58	No. 14	No. 14

Air Agitation.—The subject of air agitation has been one of careful investigation for some 10 or 15 years. It involves the chemistry of the water to be frozen, the efficiency of the filter process, the cost of power for agitation, the extra cost of the tank room work and the losses due to checking and chattering of the frozen cake. The appearance of the cake of ice is given much consideration so that the primary objective is a crystal cake with the least core or color.

The result has been designs of various sorts. The first system of agitation used involved a drop pipe into the center of each can about three-quarters of the way to the bottom of the can. Air at about 3 lb. pressure is applied and about 0.5 cu. ft. of free air per 300-lb. can is used. At a certain period in the freezing process the drop pipe is removed and the freezing is continued without agitation. Another method on the removal of the drop pipe is to suck the core and replace the water with distilled water, but this is not used in new plants at present, except in

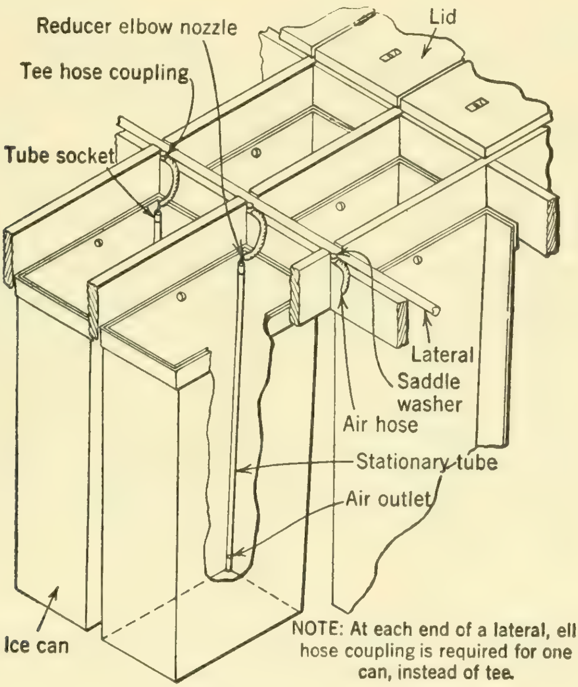


Fig. 252.—High Pressure Air Agitation.

very unusual cases. With this system, unless the drop pipe is removed at the proper time it will be frozen into the cake (requiring the thawing out by means of a steam needle), and its usefulness under these circumstances would be decreased because the pressure of 3 lb. is not great enough to prevent ice formation about the end of the pipe and the stopping of the agitation at that point. Finally, the can is lifted by a crane and is conveyed to the thawing platform where it is dipped into a tank of water at or about 70 deg. F. or in hot water (depending on local conditions) for an amount of time depending on special factors. When

the ice is loosened in the can it is placed on the tilting table and is permitted to slide on chutes into the daily ice storage room.

The low pressure air agitation[4] raw water system will give fairly good ice if good water is used. There is an expense due to the handling of the drop pipes, which must be removed at the proper time or be frozen into the ice, under which conditions the pipe has the additional expense of thawing by means of the needle, a process which results in added wear and tear to the drop pipe also. The low-pressure agitation drop pipe (Fig. 253) is placed in the center of the can and is *designed to be frozen*

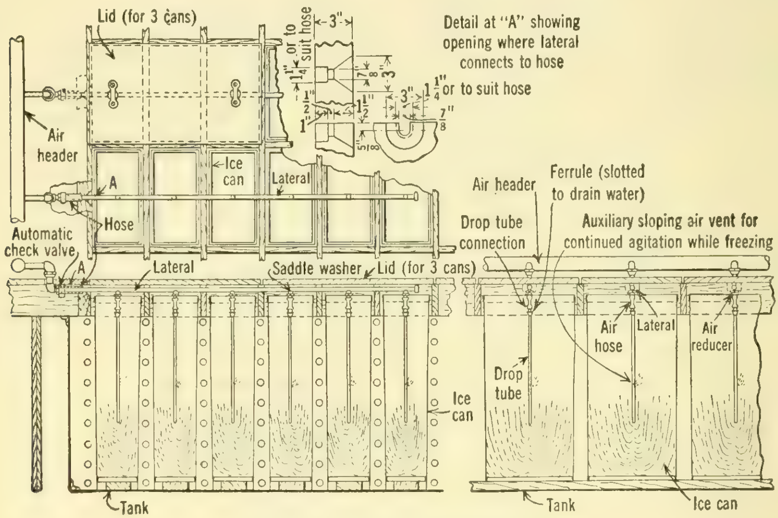


FIG. 253.—Low Pressure Air Agitation.

into the cake of ice. The seamless drawn brass tubing is perforated with small holes at several points to permit agitation of the core after the

[4] Some water causes considerable difficulty because the agitation is not vigorous enough to prevent impurities from settling to the bottom and forming a dirty streak in the ice. The cure would be either better softening (and filtration) or earlier dropping of the core, or agitation with a greater air pressure. But the low-pressure air system is much cheaper in first cost, and it is often used where the water is good. Harry Sloan says (N.A.P.R.E., 1922) that the disadvantages of low-pressure air agitation are thick opaque cores and greater labor costs. The advantages are in the low first cost, in more simple installation and most economical in the use of power.

H. R. Halterman (N.A.P.R.E., 1922) says that 100 cu. ft. of free air per minute compressed to 3 lb. per sq. in. pressure requires $2\frac{1}{2}$ hp., and 0.5 cu. ft. per minute per can is used. He states also that 100 cu. ft. of free air per minute compressed to 30 lb. requires 10 hp., but that only 0.2 cu. ft. of air per minute is required per can.

end of the tube is filled up by the formation of ice. It may be mentioned that the present tendency seems to be to permit the pipe to freeze into the cake, and thereby permit more agitation to take place, decreasing by this means the amount of white core in the finished cake.

TABLE 89

LOW-PRESSURE FREEZING SYSTEM APPARATUS—AIR BLOWERS

Tons of Ice	Number of Cans	Size of Pipe, Inches	C.f.m.	R.p.m.	Horse-power of Motor Required	Size of Blower Pulley, Inches	Belted Weight, Pounds	Maximum R.p.m. of Motor
1	16	2½	8	520	.75	7×1.5	90	1200
2	30	2½	15	635	.75	7×1.5	90	1200
3	42	2½	21	600	1.5	8×2	200	1200
4	56	2½	28	670	1.5	8×2	200	1200
5	70	2½	35	740	2	8×2	200	1800
6	84	2½	42	810	2	8×2	200	1800
8	112	2½	56	700	3	10×2.5	300	1800
10	144	2½	72	800	3	10×2.5	300	1800
12	168	3	84	505	5	12×3	400	1800
15	210	3	105	550	5	12×3	400	1200
20	280	3	140	660	5	12×3	400	1200
25	360	3	180	790	5	12×3	400	1200
30	420	4	210	460	7.5	16×4	730	1200
35	484	4	242	505	7.5	16×4	730	1200
40	560	4	280	565	7.5	16×4	730	1200
50	720	4	360	680	7.5	16×4	730	1200
60	840	4	420	460	7.5	16×4	1460	1200
80	1120	4	560	565	7.5	16×4	1460	1200
100	1440	4	720	680	7.5	16×4	1460	1200
120	1680	6	840	500	10	20×4	3000	1200
150	2160	6	1080	550	15	20×4	3000	1200
200	2880	6	1440	550	15	20×4	4500	1200

Blowers operate at 3 lb. gage pressure.

Very seldom is the water to be frozen free from some trace of solids. The saying that water which is good enough to drink is good enough for raw water ice needs some reservations (Fig. 254.) The solids concentrate

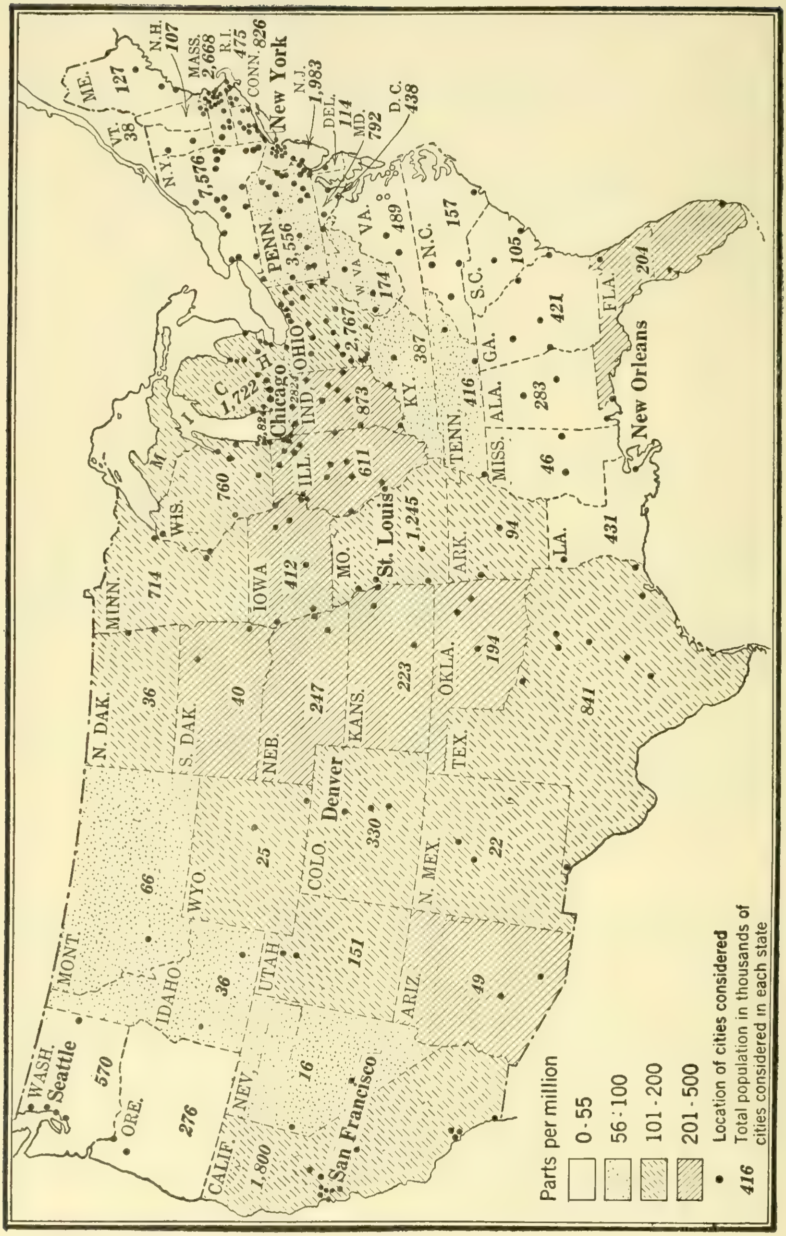


Fig. 254.—Hardness of Well Water in the United States.

as freezing continues, and the core water becomes very briny to the taste. The core needs to be removed once and sometimes twice in the 300- and 400-lb. cake, both from the viewpoint of the foreign material and also because of the reduced freezing temperature of the core water and the increased time of freezing required. This is done by means of a core sucker (Fig. 265). The air for agitation is best taken from the top of the tank, thereby getting a colder air than would be available in the summer from the atmosphere and a lower moisture content. The air is delivered, as a rule, by means of large headers to the laterals. The air pressure is therefore more uniform all over the tank.

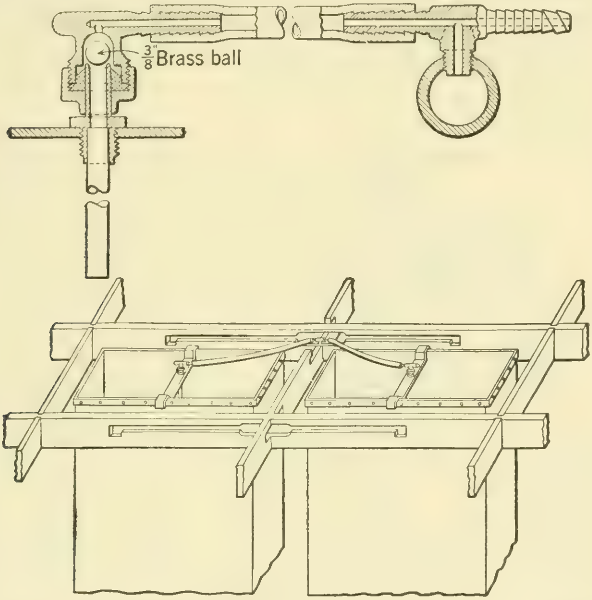


Fig. 255.—Medium Pressure Can Agitation—Center Tubes

The High and the Medium Air Pressure.—If the drop pipe is located in the center of the can it is subjected to 32 deg. F. until the very last part of the freezing of the can, and under these conditions no trouble is experienced from the freezing of the moisture content of the air. However, if the drop pipe is made an integral part of the can, either soldered in the corner (Fig. 252) or fastened to the side of the can, it comes into metal contact with the brine, which is at 14 to 18 deg. F. Such air agitation, spoken of as high pressure or medium pressure air, will need to be dehumidified in order to reduce the dew point of this air. As air under 10 to 20 lb. is required at the laterals, the compressor usually

supplies air at a pressure greater than this by an amount varying from a few pounds up to 30 lb. more than is required at the cans.[5] Some manufacturers have no high pressure, but what is called a medium pressure air system (Fig. 255). In another form the drop pipe is soldered into the corner of the can, and a hole in the side of the pipe about 6 in. from the bottom of the can permits the air to come out. However, dehumidifiers are required in all cases except the low-pressure systems, and there are three forms of these in general use.

The Dehumidifier.—The first form employs a cylindrical shell one-half full of water and kept cool with brine from the brine tank. The air is admitted near the bottom by passing through pipes or plates perforated with small holes, and as it passes upwards through the water, is washed, cooled and purified. Apparently much depends on the design of the air openings as it is possible for the air to pass in streams instead

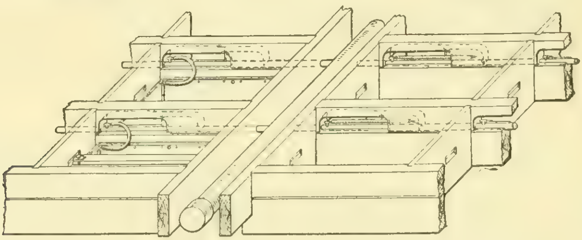


FIG. 256.—Medium Pressure Can Agitation—Corner Tubes.

of as bubbles under which conditions the efficiency of the dehumidifier will decrease. Finally, the air is further cooled and dehydrated by means of a brine coil.

The second type uses two cylindrical shells through which the air passes in series, the first being one-half full of water cooled by means of a brine coil and the second half-full of brine which is cooled by the use of direct-expansion piping. The air line to the cans has also usually

[5] For 18 lb. air a hole made by a No. 75 drill is satisfactory or a No. 53 drill and a No. 1f B. and S. gage wire. The air should be compressed to 25 to 30 lb. at the compressor if 14 to 16 deg. F. brine is to be used.

The high-pressure air system has the advantage because of the necessity of the removal of the air tube in the low-pressure system, with the resulting stoppage of agitation and the large core in the ice block as well as the labor required in the work. The high-pressure air system requires the use of a constricting orifice.

Thomas Shipley says that the low-pressure air system is all right with proper care and some waters, but the high-pressure air system would produce much better results on the average than will the low-pressure even when the low-pressure agitation is doing its best as there is no stoppage of agitation at any time; and that the core spaces do not require washing out except when the sediment is very heavy.

a cartridge filter which has to be recharged every four hours. A modification of the method uses a brine spray (Fig. 257).

The third type is composed of two shell and coil or shell and tube cylindrical tanks, brine cooled,[6] through which the air passes in series. The second tank collects a frost on the cooling surfaces and must be defrosted every six to eight hours.[7] This is done by the simple device of reversing a four-way valve. The particular advantage of this type is its simplicity, the lack of care required in its operation, the avoidance of dilution of the brine as is the case where the air comes in direct contact with the brine and the lack of any need of filtering in order to collect the salt held in the air which would tend to close the needle valves in the can pipes. The frost accumulation on the pipes assists in cooling the air by an amount at least equal to the latent heat of fusion of the ice when the four-way valve is reversed (Fig. 258).

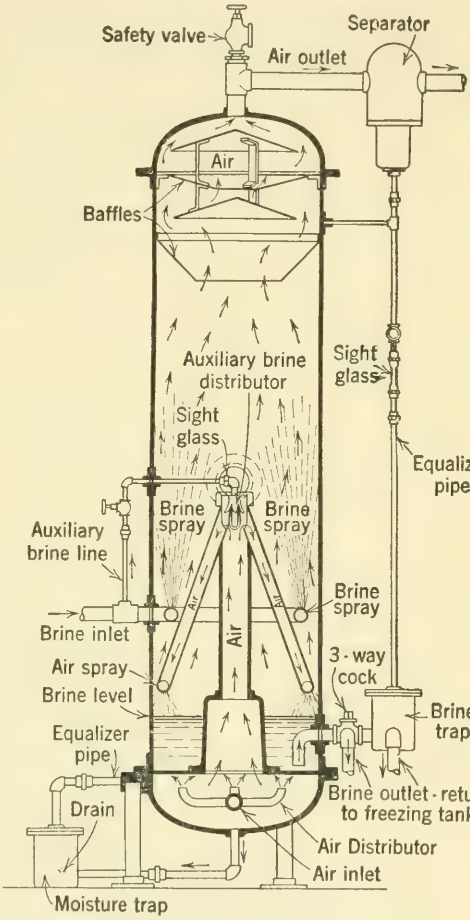


Fig. 257.—Air Dehydrator, Using Brine Sprays.

The use of air agitation involves considerable operating expense

[6] Shipley says that the use of brine instead of direct expansion for air cooling is preferred because if the compressor stops at any time the air can be kept cold and therefore dehumidified. Air pressures of 5 lb. for low pressure and 25 lb. for high pressure use the same motor for either condition and therefore the same power.

[7] The possible objection to this is the accumulation of frost on the second brine coil which soon decreases the efficiency of the cooling surface to a point where cooling below 32 deg. F. would be difficult, unless the four-way valve is operated as well as the brine valves.

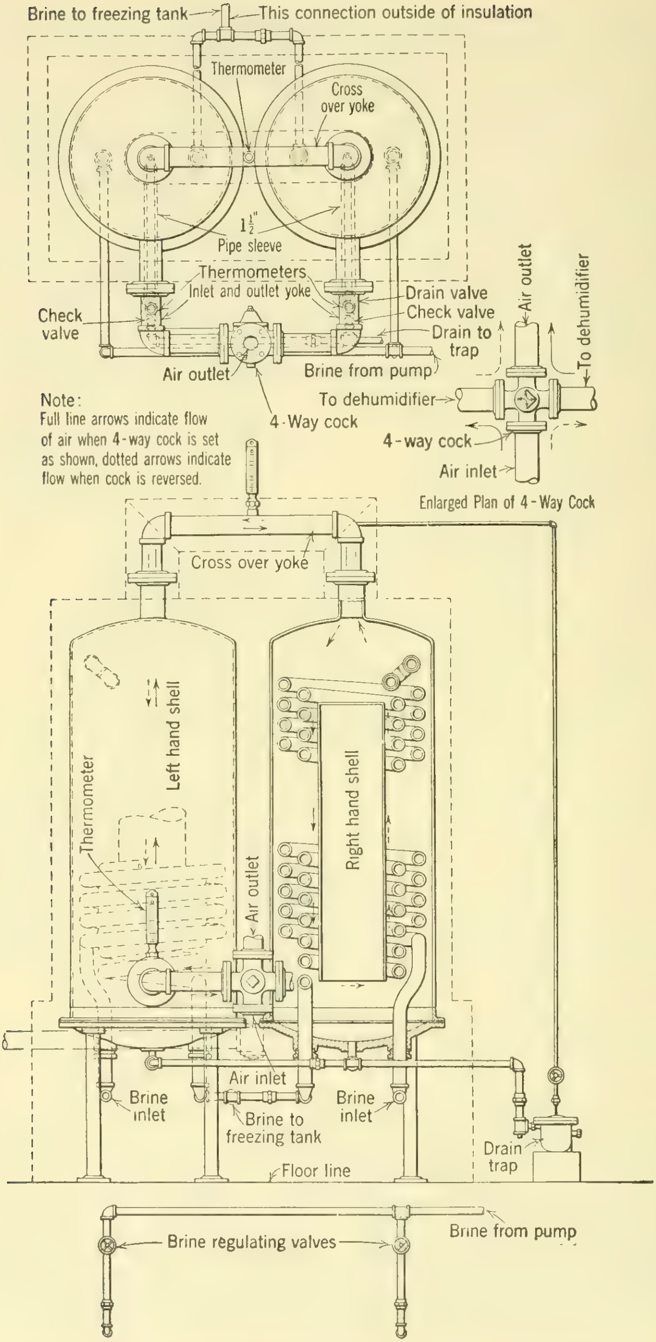


FIG. 258.—Air Dehydrator—Using Brine Coils.

and investment. The medium air pressure system requires slightly less air than the low-pressure agitation and a little more power, but the ice quality is better. It is claimed that the high agitation[8] system requires 6 kw. hr. per ton, or approximately one-eighth of the entire power required of the plant. One man is used in the tank room per shift up to 60 tons output per 24 hours. The ratio of the power requirements is about 1.0 hp. per 100 300-lb. cans for low-pressure air, and from 4.0 to 5.0 hp. per 100 300-lb. cans for high-pressure air agitation, and in addition there is required the refrigeration necessary to cool the air, and the power for the water and the brine pumps. About $\frac{1}{2}$ hp. is used per 100 300-lb. cans for the core pump. If air agitation could be eliminated, it would be a great means of simplifying the manufacture of ice, and attempts are being made in that direction. The modern tendency (1927) is towards the use of a medium-pressure air (about 10 lb.) using a drop pipe in the center of the can, made of brass and extending nearly to the bottom of the can (usually about 5 in. of the bottom) as compared with about 9 in. for the low-pressure system. It gives a cake of ice with hardly any perceptible core, saves tank room labor and does not require much more power than the rather crude low-pressure drop pipe system of the earlier designs.

Ice Dumping and Refilling the Cans.—The tank room routine consists in the filling of the cans with water, the freezing and, finally, the dumping of the ice. The usual method is to lift a number of cans by means of the can hoist and to convey them to the dump where the can filler is usually located. Much of this work is heavy routine, and it takes almost as long to lift two cans as it does to lift six or eight. To reduce the cost of labor in the tank room 2-can hoists are now limited as a rule to very small ice tanks, but the larger tanks have cranes which will lift as many as 24 to 30 cans at a time, using the heavy 3-motor crane. Heretofore the connection from the cans to the crane has been by means of can dogs, but when four, six or more cans were "lifted" considerable time was wasted because of the undue swinging of the cans at the start and the stop, and injury to the cans or the direct expansion piping in the tank. The result has been the use of the so-called can basket—a device which means simply the assembling together of all the cans to be lifted at one time, and a means of attaching the crane chains to the basket. Suitably held together a row of 24 cans can be manipulated nearly as quickly as the old 2-can hoist. The basket is

[8] Sloan says (N.A.P.R.E., 1922): Every hp. additional added to auxiliaries of the plant mean $\frac{1}{4}$ kw.-hr. per ton ice (allowing for efficiency of motors, etc.). Whyte says in reply that cost of power in five plants (Consumers Ice Co., Chicago) checked exactly with amount used in low-pressure plants.

sometimes made of an angle iron frame, and the additional weight assists in submerging the cans in the brine and also decreases the necessary strength of the tank framing, and permits the cans to be placed on closer centers. In some designs the use of baskets requires, however, the placing of the piping on the *bottom of the tank* or its elimination entirely by the use of the shell and tube brine cooler. The saving of time in the tank room has been found to be about 30 per cent using baskets as compared with can dogs. With the decreased size of the tank frame by using the basket it has been found that less tank room, roof and crane space are required, less ground space, insulation and foundation work, less framework and brine charge, fewer accidents and less

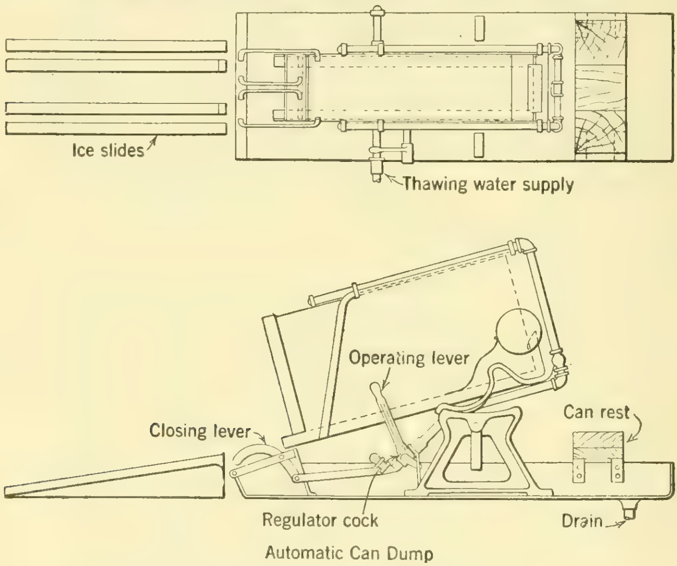


Fig. 259.—Automatic Can Dump.

damage to the cans and piping, etc., resulting in 10 per cent less space and 50 per cent less repairs (Ralph Cole, N.A.P.R.E., 1923).

Tests of ice tanks (R. C. Doremus, N.A.P.R.E., November, 1922) showed considerable variation in the operating conditions when using large baskets. With 960 400-lb. cans, using $38\frac{1}{2}$ diam. by 14 ft. long shell and tube brine cooler and 2 18-in. propellers at each end, pulls of 12 cans caused a brine temperature rise of 0.7 deg. F., whereas 16 cans at a time caused a temperature rise of 1.0; 20 cans, 1.6, and 24 cans caused a temperature rise of 1.9 deg. F. Lifting 8 cans at a time caused the brine level to fall $1\frac{1}{4}$ in., and 24 cans caused the brine level to drop $5\frac{1}{4}$ in. These changes of brine level would require special designs. It is

claimed that a can lift of 1 to $1\frac{1}{4}$ per cent is the maximum amount permissible at a time unless an excessive variation of the operating conditions is to be permitted.

The Dip Tank.—The old-fashioned sprinkler dump (Fig. 259) is now being replaced by the dip tank. One of the operating difficulties in ice making is that of the checking of the ice. Checking is due to so

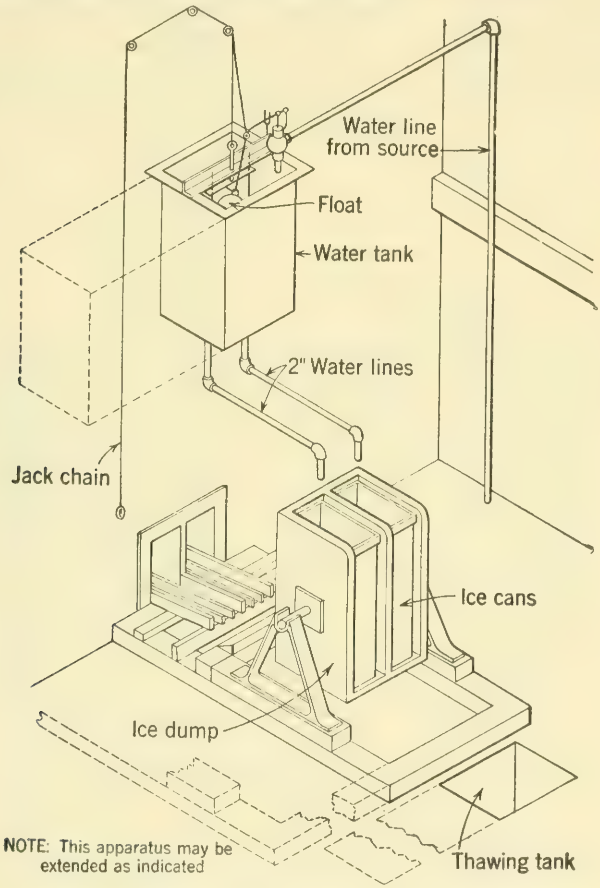


FIG. 260.—Automatic Can Filler.

many factors that operation in any two plants is seldom alike. Some plants use 60 deg. F. water in the dip tank and others have the water steaming. The present tendency is to have the dip tank water agitated with air (in the case of medium and high-pressure systems) in order to give greater heat transfer to the cans and to stimulate a more even temperature all around the cake of ice, through destroying stratification

by removing the cold film of water about the cans. A gain of 42 per cent in the time required for thawing of the ice is claimed for the air-agitated dip tank.

Can Filling.—Even in the small ice-making plants it is doubtful whether the old float valve type of automatic can filler is profitable. It is slow, expensive to operate, and unless carefully operated may result in a weakened brine. At present the dip tank, the can dump (which is simply a rest for the cans on trunnions) and the measuring tank type of can filler are designed as a unit (Fig. 260). With the modern type of can filler the correct amount of water is ready in each compartment (there is

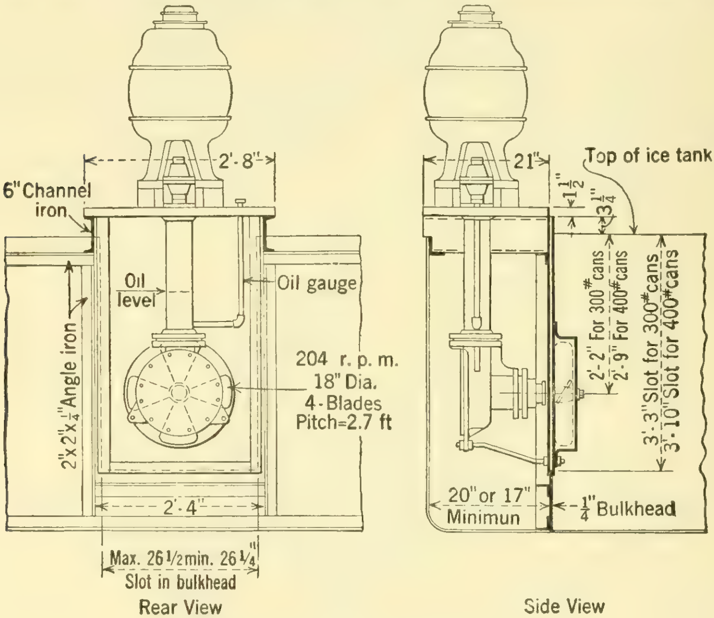


FIG. 261.—Vertical Brine Agitator.

a separate compartment for each can) and a pull of the control chain starts the filling operation. With a 3-in. filler pipe the 400-lb. can can be filled in 26 seconds, whether 2 or 20 cans are filled at the same time. While the tank man is pulling another batch of cans the filler tank is receiving another charge of water.

The cans, filled with water, will submerge quickly to the proper level without difficulty and with less wear and tear to the equipment. This is especially important of late years because of the amount of agitation given the brine. This would tend to carry the can out of plumb unless they were restrained by the means of can guides. The dump, for the

larger plants, is electrically operated by means of cables driven by a slow speed electric dump hoist.

Agitation.—The brine must be agitated (Figs. 261 and 262 and Table 90) in order to secure results of any value in the ice tank. This is because of the tendency for stratification and the formation of films about the ammonia piping and the ice cans. The advent of raw water ice has

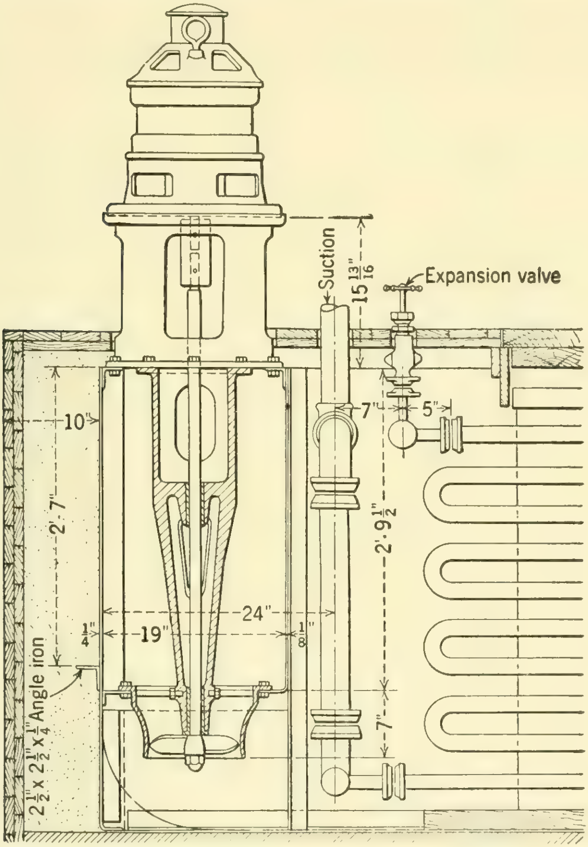


FIG. 262.—Vertical Brine Agitator.

changed conditions considerably in this respect. Raw water ice can be made, as a rule, only with 14 deg. F. brine or higher, because of the tendency for checking. It is also uneconomical to operate an electrically driven compressor with low-temperature brine. The net result is a striving for a close temperature range between the temperature of the ammonia boiling in the pipes and of the brine.

TABLE 90

STANDARD FREEZING TANK PROPELLERS

Capacity, Tons	Horizontal										Vertical					
							Motor		Motor pulley						Motor	
	Number	Diameter, inches	Pitch, inches	Revolutions	Rubber belt, inches	Pulley, inches	Horsepower	R.p.m.	Diameter, inches	Face, inches	Number	Diameter, inches	Pitch, inches	Revolutions	Horsepower	R.p.m.
1	1	12	6.0	103	3	34					1	7	6	850		
2	1	12	6.0	125	3	28	1	1160	3	$3\frac{1}{2}$	1	7	6	850	2	850
3	1	12	6.0	145	3	24					1	7	6	850		
4	1	15	12.0	145	$3\frac{1}{2}$	32					1	9	6	850		
5	1	15	12.0	165	$3\frac{1}{2}$	28	2	1160	4	$4\frac{1}{2}$	1	9	6	850	3	850
6	1	15	12.0	180	$3\frac{1}{2}$	26					1	9	6	850		
8	1	15	22.0	145	4	32					1	9	9	850		
10	1	15	22.0	165	4	28	3	1160	4	$4\frac{1}{2}$	1	9	9	850	3	850
12	1	15	22.0	180	4	26					1	9	9	850		
15	1	15	22.0	193	4	30					1	12	12	850		
20	1	22	39.9	152	$4\frac{1}{2}$	38	5	1160	5	$5\frac{1}{2}$	1	12	12	850	5	850
25	1	22	39.9	160	$4\frac{1}{2}$	36					1	12	12	850		
30	2	22	39.9	130	5	36					2	9	9	850		
35	2	22	39.9	153	5	38	$7\frac{1}{2}$	1160	4 5	$5\frac{1}{2}$	2	9	9	850	3	850
40	2	22	39.9	153	5	38					2	12	12	850		
50	2	22	39.9	180	6	38	10	1160	6	$6\frac{1}{2}$	2	12	12	850	5	850
60	2	22	39.9	180	6	38					2	12	12	850		

In order to get this close operation and at the same time to acquire production a heavy agitation is necessary. Some tanks have been designed for a difference in the brine level of from three to five or more inches in the tank. There is also more thought in regard to the regulation of the velocity of the brine in different parts of the tank. With a tapering can it is evident that there is more space at the bottom of the brine tank between the cans for the brine to pass through than there is at the top. Also there is usually considerable space at the bottom of the tank below the cans. Since for best results the brine flow should be reasonably uniform around the cans, attention is now being directed to this matter.

The amount of agitation is not at all uniform, as has been mentioned, but the following is the practice among some engineers: for 400 cans or less use one agitator, and for 24 cans wide and about 20 cans up to 35 cans long use two agitators, at the same end of the tank. For more than 24 cans wide and more than 35 cans long use *two* agitators at *each* end. The power required of the agitators is approximately $2\frac{1}{4}$ hp. per agitator, propeller 204 r.p.m., 18 in. diam., 4 blades, 2.7 ft. pitch.

Water Softeners.—*Deep well water* varies in the amount of salt in solution in the different parts of the United States (Fig. 254). The eastern seaboard has good water except in Florida, where it is very bad. The central states have hard water, as for example, Illinois, Indiana, Iowa, South Dakota, Nebraska, Kansas and Oklahoma. Washington and Oregon have soft water, but California water is bad. The water of the Great Lakes is hard. In the southwest the water in certain regions contains so much sodium bicarbonate as to make it unfit for raw water ice making.

If water contains more than a trace of solids in solution before it is suitable for can ice manufacture it must have a special treatment if a crystal cake of ice is to be secured, for as freezing continues the mineral impurities are separated out as a precipitate and will show as a white or as a colored deposit. Thus magnesium salts will cause a white and iron salts a red deposit in the ice. If agitation is not resorted to these particles as well as the air in solution in the water will settle on the surface of the ice and being frozen into the ice will give the peculiar opaque appearance which is not desired by the retail trade. If the water is agitated, the impurities will be kept moving in the core water, and this action will continue until the core water gets more and more briny until settlement will occur or the core is removed, to be replaced with fresh water (or under exceptional circumstances with distilled water). If the core was a large one requiring a large replacement, it is possible that a second core pulling would be required. In addition to

Calcium carbonate is limestone ($CaCO_3$). Water (H_2O) as rain, absorbs carbon dioxide (CO_2) as it falls to earth, and from the earth, and then the water with the absorbed CO_2 dissolves the limestone. The chemical equation is written as follows:

$$CaCO_3 + CO_2 + H_2O = CaCO_3 \cdot CO_2 \cdot H_2O$$

Limestone + Carbon + Water = Calcium Bi-carbonate
(Insoluble) Dioxide (Soluble)

The same is true of magnesium carbonate. But water chemists have reported the bi-carbonates of calcium and magnesium in terms of calcium carbonate and magnesium carbonate for so long that it has become standard practice. Occasionally, however, an analysis is submitted where they are expressed as bi-carbonates, and the following factors may be of help:

Calcium Bi-carbonate × 0.617 = Calcium Carbonate.
Calcium Carbonate × 1.62 = Calcium Bi-carbonate.
Magnesium Bi-carbonate × 0.576 = Magnesium Carbonate.
Magnesium Carbonate × 1.735 = Magnesium Bi-carbonate.

The following table gives the names of the usual minerals found in water, how they effect the appearance of raw water ice, and the effect of treatment with hydrated lime in an efficient water treating plant.

Minerals in Water	Effect in Ice	Result of Treatment With Hydrated Lime $Ca(OH)_2$
Calcium carbonate	Forms gritty, dirty, discolored deposit, usually in lower part and center of the cake. Causes shattering at low freezing temperatures.	Practically eliminated.
Magnesium carbonate.....	Forms gritty, dirty, discolored deposit, milky patches and bubbles, and also causes shattering at low freezing temperatures.	Practically eliminated.
Iron oxide.......	Causes bad discoloration, yellow or brown deposits and also stains the calcium and magnesium deposits.	Eliminated.
Aluminum oxide and silica......	Cause dirty deposit and sediment.	Practically eliminated.
Suspended matter	Causes dirty deposit and sediment.	Eliminated.
Calcium sulphate calcium chloride	Act like and are no worse than sodium sulphate and sodium chloride. Do not form deposit.	No change.
Magnesium sulphate.......	Cause greenish or grayish cast, concentrate in core water, retard freezing and cause heavy cores. Often show up as white spots and dirty colored streaks. Also act like sodium sulphate and sodium chloride. Do not form deposit.	Changes to calcium sulphate.
Magnesium chloride.......		Changes to calcium chloride.
Sodium sulphate, sodium chloride.	Cause white butts, concentrate in core, make heavy core and retard freezing—but do not form deposit.	No change.
Sodium carbonate (actually present as sodium bi-carbonate) . .	In only small quantities often causes shattering at temperatures below 16 degrees. Also causes white butts, concentrates in core, retards freezing and makes heavy cores. Does not form deposit.	Treatment changes sodium bi-carbonate to sodium carbonate —treatment improves but little.

Minerals	Allowable Limits (With Remarks)
Calcium carbonate and magnesium carbonate (combined)	If 70 p.p.m. (4 grs. per gallon)—or less—lime treatment required but *alum* treatment recommended to change the carbonates to less objectionable calcium sulphate and magnesium sulphate. If total exceeds 70 p.p.m. (4 grs. per gallon)—lime treatment must be used to prevent objectionable deposit. Calcium and magnesium carbonates can be present in any quantity—lime treatment will remove all but 30 or 40 p.p.m. (1.8 or 2.4 grs. per gallon).
Iron oxide	If more than 0.2 p.p.m. (0.012 gr. per gallon) treatment is required. This trace of iron may be sufficient to badly discolor deposits of calcium and magnesium that otherwise might be unobjectionable. Some analyses report the iron and aluminum oxides together. Such analyses are valueless because iron is so objectionable that its quantity and form should be fully understood.
Aluminum oxide and silica	Not usually present in quantities which alone would require treatment.
Suspended matter	If only objectionable impurity in the water, filtration is the only treatment required.
Color	Analyses may indicate the presence of color, but not show to what color is due. Surface waters, low in dissolved mineral matter, but with color, should be given expert individual consideration, as color may or may not be easily removed.
Calcium sulphate Magnesium sulphate Calcium chloride Magnesium chloride Sodium sulphate Sodium chloride Sodium carbonate	All the sulphates, chlorides and the sodium carbonate have, in general, the same effect on the appearance of raw water ice and, therefore, in this rough tabulation, can be grouped together. Generally, the sodium carbonate can be considered $1\frac{1}{4}$ times as objectionable as the sulphates, and the chlorides about $\frac{3}{4}$ as objectionable as the sulphates. Therefore total the sulphates and chlorides separately and apply the factors: Total sulphates $\times$ 1.0 = ... Total chlorides $\times$ 0.75 = ... Sodium carbonate $\times$ 1.25 = ... Sum Total = ... If sum total is less than 171 p.p.m. (10 grs. per gallon) first quality ice can be expected with efficient air agitation and average brine temperatures. If sum total is more than 171 p.p.m. (10 grs. per gallon) and less than 256 p.p.m. (15 grs. per gallon)—good ice can be expected with slightly higher than average brine temperatures or slightly more than the usual volume of air for agitation. If sum total is more than 256 p.p.m. (15 grs. per gallon) and less than 342 p.p.m. (20 grs. per gallon)—merchantable ice can be made if much more than the usual volume of air per can per minute is provided, especially during the beginning of the freeze and brine temperatures of 18 degrees or more are carried. If sum total is more than 342 p.p.m. (20 grs. per gallon) and less than 684 p.p.m. (40 grs. per gallon)—the analysis or another sample of water should be submitted, together with full details of operating conditions, for a complete report as to whether an acceptable quality of ice can be made. If sum total is more than 684 p.p.m. (40 grs. per gallon)—merchantable ice cannot be expected.

Name	Symbol	Reacts With	Becomes
Calcium bicarbonate......	$(CaH_2)(CO_3)_2$	$Ca(OH)_2$	$\underline{2CaCO_3} + 2H_2O$
Calcium sulphate.........	$CaSO_4$	Na_2CO_3	$\underline{CaCO_3} + Na_2SO_4$
Calcium chloride.........	$CaCl_2$	Na_2CO_3	$\underline{CaCO_3} + 2NaCl$
Magnesium bicarbonate...	$MgH_2(CO_3)_2$	$Ca(OH)_2$	$\underline{MgCO_3} + Na_2SO_4$
Magnesium sulphate......	$MgSO_4$	$Ca(OH)_2$ and Na_2CO_3	$MgSO + Na_2CO_3 = MgCO_3 + Na_2SO_4$ $MgCO_3 + Ca(OH)_2 = \underline{Mg(OH)_2} + \underline{CaCO_3}$
Magnesium chloride......	$MgCl_2$	$Ca(OH)_2$ and Na_2CO_3	$MgCl_2 + Na_2CO_3 = MgCO_3 + 2NaCl$ $MgCO_3 + Ca(OH)_2 = \underline{Mg(OH)_2} + \underline{CaCO_3}$
Iron bicarbonate.........	$FeH_2(CO_3)_2$	$Ca(OH)_2$	$FeH_2(CO_3)_2 + Ca(OH)_2 = \underline{FeO} + CO_2 + \underline{CaCO_3} + 2H_2O$ $\underline{2FeO} + O_2 = \underline{Fe_2O_3}$

The underlined substance is insoluble.

the poor appearance of these salts in the ice, their presence has the effect of increasing most decidedly the amount of checking of the ice, resulting in a direct loss unless there is a market for crushed ice.

The treatment for the softening of water is quite simple. Certain salts cannot be removed, as, for example, the sodium compounds, whereas the calcium and magnesium salts usually can be precipitated. The usual method is to use lime ($Ca[OH]_2$) or lime and soda ash (Na_2CO_3). If the sum of the sulphates, chlorides and nitrates plus the sodium carbonate does not exceed 20 grains per gal. the water can be purified for ice making. From 20 to 40 grains it is doubtful, and over 40 grains per gal. gives water that is not satisfactory for ice making at all and may not be used for raw water ice making.

Up to 12 grains per gallon of sulphates, chlorides, nitrates and sodium carbonates (altogether) the low-pressure air system will answer. If high-pressure air is used at the bottom of the can this sum may be increased to 20 grains and the ice will look as well as 12-grain per gallon water with low-pressure agitation.

With 0.02 grain "iron oxide" per gallon removal is necessary or color will appear in the ice, especially as the ice appears to magnify any foreign material in the ice itself. It has been found that sodium salts particularly require agitation at the bottom of the can.

Treating the water always decreases the amount of core pumping necessary, and reduces in proportion the care required of the attendant on the can floor. The core becomes more and more concentrated as freezing continues. Calcium and magnesium carbonate freeze into the ice earlier than any other salts, being precipitated because of the loss of the carbon dioxide gas which held them in solution as bicarbonates, and this is hard to pump out unless a large core is pumped. These carbonates if frozen into the ice will remain in the ice box when the ice is melted.

Generally speaking, mineral matter begins to freeze into the ice when the concentration becomes from 55 to 60 grains per gallon. Ice made from hard water is not so firm, hard and crystal-like as if the water were treated, but is likely to be brittle and is far more likely to shatter. The presence of bicarbonates results in small white specks or bubbles frozen into the ice.

Certain water requires distillation, and water heavy in sodium sulphate, sodium carbonate, etc., is likely to have a white shell form around the bottom of the cake.

The treatment with lime and soda ash necessarily *takes place cold*, under which conditions the reactions take place slowly and incompletely. The usual method is to permit the water and the chemical to stand for four hours in the water softener settling tank. During the sedimentation

period the chemicals sink to the bottom where they are drained off as a thin sludge, and the softened water is drawn off near the top and is passed through a quartz filter to remove the free iron, silica and the suspended dirt and sludge.

The tendency of the design of water softeners is to give more time to sedimentation, about 6 to 8 hours in fact, thus securing more complete chemical reaction and settling. The filter is being made larger and is figured on the basis of a maximum of 3 gal. per sq. ft. per minute of cross-sectional area, and 12 gal. per min. per sq. ft. for backwash. There is also an alum coagulator to improve the filter action, although some plants add the alum in the downtake of the softener.

The muddy appearance of the ice in the spring and autumn may be caused by dirty water or it may be by the intermittent action of the filters during periods of light loads. The remedy is to recirculate the water and run at higher capacity.

Water Softening for Can Ice Making.[9]—"The requisites for first-quality ice are clearness, firmness and freedom from discoloration. These qualities are possessed by ice made from dissolved solids and gases, such as the reboiled distilled water which has, until comparatively recently, been almost exclusively used in the artificial ice industry. Ice frozen from impure water is opaque, discolored, or brittle, depending on the nature of the impurities.

"Freezing water is, in many respects, much like boiling and evaporating it, in that by far the greatest part of the substances dissolved in the water freeze out in the ice made from it. The most effective purification of raw water for ice making is, therefore, that which reduces the objectionable impurities in the water to a minimum. It is now generally recognized that the most effective purification is accomplished by lime-soda softening, followed by sand filtration.

"In the process of the manufacture of ice from raw water, cans of the water are surrounded by the sodium or calcium chloride brine having a temperature of 12 to 18 deg. F. Air under either high pressure (15 to 25 lb.) or low pressure (3 to 5 lb.) is bubbled through the water as it freezes, the high-pressure air being in general more effective. The first ice formed around the sides of the can is usually relatively pure. The dissolved gaseous and solid impurities in the water are frozen and begin to deposit on the face of the ice; but the currents of water set up by the air agitation wash these impurities off the surface of the ice and carry them towards the center of the can. The impurities in the raw water thus become concentrated in the unfrozen water in the middle of the can. If these impurities are insoluble, their accumulation in this unfrozen water usually becomes so heavy that eventually the currents of water set up by the air agitation are not powerful enough to keep the particles in suspension. As a result these white or colored particles are deposited in the ice before the cake is frozen solid, or if the impurities are soluble, as, for instance, sodium salts, their concentration may become so great that freezing is materially retarded. In either case this concentrated impure water, or 'core water' as it is termed, is generally removed, usually with a suction pump, and is replaced with fresh water. The solids and gases left in the core water, or introduced in the fresh

[9] A. S. Behrman. Journal of Ind. and Eng. Chemistry, March, 1921.

water replacing it, appear as white or colored deposits, and as air needles in the core of the ice when the cake is frozen solid.

"The most common, and at the same time the most undesirable class of calcium and magnesium compounds are those causing temporary hardness—that is, the bicarbonates. Just as heating a water of this nature causes precipitation of the normal carbonates, so will freezing it drive off the loosely held 'half-bound' carbon dioxide and cause a precipitation in the ice of the normal carbonates, and possibly of magnesium hydrate. With air agitation these precipitated compounds will be carried more or less completely to the center of the can. Here they will accumulate until it becomes necessary to pump out the heavily laden water and replace it with fresh water. Frequently this accumulation takes place so rapidly that two, and sometimes even three, core pumpings are required. Even with good air agitation, however, the removal of the precipitated compounds to the middle of the can is often not complete, and milky white dots, bubbles and patches are found distributed throughout the clear portion of the ice. Frequently also a white opaque shell of the precipitated carbonates will be found around the lower portion of the cake, where freezing is most rapid. When ice containing the precipitated carbonates melts it leaves this objectionable sediment.

"Softening with lime removes the bicarbonates effectively and cheaply, and leaves in the treated water no products of the reaction beyond the 2.5 to 4 grains per gallon of calcium carbonate and magnesium hydrate generally considered the limit of the lime reaction in the cold.

"The removal of permanent hardness is far less important for ice making than of temporary hardness. Investigations now under way indicate that in a great many cases, possibly all permanent hardness need not be removed, provided that the magnesium which always tends to make white ice is removed from such compounds and is replaced with calcium. This is accomplished of course in the treatment with lime. The calcium sulphate and chloride left in the water as a result of the lime treatment appear to be no more detrimental—and in some cases even less so—than the sodium salts which would result in the removal of the permanent hardness with soda ash. In practically every case ice made from the plain lime treatment is equally as good as or better than when soda is used. In addition, sedimentation in the softening tanks is more complete, thereby reducing the load on the filters. Further, when soda ash is used the carbonate ions in the treated water are lessened and as the ice freezes a much greater concentration in the unfrozen water is required before the ion-product constant is exceeded. As a result the unfrozen water remains clear much longer, and free from particles of the precipitated carbonate that would tend to deposit on the ice, consequently core pumping can be delayed, a less amount of water can be used and the refrigeration thus otherwise wasted, can be saved.

"As little as 0.2 parts per million of iron may cause 'red ice,' that is ice colored reddish brown, principally in the core. Silica and alumina are deposited in the core of the ice cake, imparting a muddy appearance to it, and when this ice melts there remains a slimy sediment. Organic matter is frequently found in objectionable quantity in surface waters, particularly in warm weather. It usually colors the core of the ice a muddy or bright yellow, which is sometimes so objectionable that the ice is saleable with difficulty although it is of good quality in other respects. Lime-soda softening of the raw water followed with sand filtration aided by the use of a coagulant effectively removes the iron, reduces the silica and the alumina usually by from one half to three quarters and greatly lessens the amount of objectionable organic matter. Sometimes bleaching powder can be used to advantage

to remove organic matter. The chief objection to sodium salts (and potassium) is that they accumulate in the core water and retard freezing and are finally deposited as white solids in the ice. If a considerable amount of sodium salts is present the lengthening of the freezing period may be so serious as to require several core pumpings and refillings with fresh water. In addition to this general objection certain sodium compounds have specific undesirable results. Sodium bicarbonate in considerable amount tends to cause brittleness and cracking. Large quantities of sodium sulphate tend towards the formation of a white shell on the outside of the cake, giving the whole cake the appearance of marble even though the interior portion is quite clear. Treatment with lime converts the bicarbonate of soda to the normal carbonate and decreases somewhat the tendency towards brittleness. Softening has no other beneficial effects, as the only way to remove sodium salts is to apply distillation.

"**Checking and Cracking.**—There is a tendency in ice making for the ice to crack and shatter, especially when a low temperature brine is used, and the only explanation that can be offered is that the ice is frozen under some internal strain. It is quite possible that the presence of bicarbonates is chiefly responsible for this strain. During the freezing process, while the half-bound carbon dioxide is trying to escape, the ice continues to crystallize, entrapping bubbles of gas and particles of the precipitated compound which are readily visible. The ice thus formed is comparable to a metal casting full of blow holes and impurities and is in consequence inherently weak and brittle.

"Some weight is given this hypothesis by the general experience that removing the bicarbonates of calcium and magnesium from the water by treatment with lime results in the production of much clearer and firmer ice, and frequently permits the use of lower brine temperatures. Further in a recent series of experiments ice was frozen from water to which had been added varying amounts of sodium bicarbonate. In all cases except the lowest concentration (10 grains per gallon) the ice formed was quite brittle, cracked readily, and showed evidence of a bubbly structure. Analysis of the melted core ice showed the conversion of the bicarbonates to the normal carbonate in all cases to the extent that the normal carbonate alkalinity averaged 35 per cent of the bicarbonate alkalinity.

"Finally the question arises as to the limiting quantities of the various impurities that a raw water can carry, and still make first class ice. This is not known in all cases. Obviously in the cases of the bicarbonates of calcium magnesium and iron, the limiting concentrations are their own solubilities, since softening with lime leaves the same residual content regardless of the original concentration. It is also probable that the permissible maximum of silica and alumina is not exceeded in natural waters if treated with lime.

"With regard to sodium salts, and to calcium sulphate and chloride, investigations tend to indicate that when the total soluble salt content of a raw water exceeds 30 to 40 grains per gallon exclusive of the temporary hardness, first quality raw water ice cannot be made even with water softening high-pressure air agitation."

Checking and Cracking of Ice.—The advent of raw water ice making has brought many problems, and one of these is that of cracking and checking of the ice cake during freezing, or during the pulling period or even up to the period of storing in the daily ice storage room. The

causes of checking are many, and a few of the principal ones will be given in the following:[9a]

1. Poor brine circulation in the brine tank.
2. Allowing the cans to temper while standing in draughts of warm fresh air, especially if the sprinkler system is used, though tempering will be found to be good for the ice in some cases if the cake is not subject to draughts.
3. Filling the cans with water that is too warm.
4. Rough handling of cans by tank men during core sucking, hoisting and dumping.
5. Lowering cans into the thawing tank too suddenly, or using a thawing water of too high a temperature.
6. Leaky cans.
7. Too low a brine temperature (approximately 44 hr. is the correct freezing time for 300 lb. cans).
8. Presence of bulged cans.
9. Brine level below the level of the water in the cans.
10. Dissolved calcium carbonate in the water to be frozen, especially when calcium sulphate and sodium sulphate are present.

From the preceding list it is clear that the causes of checking are not clear and definite, and therefore the correction of this trouble is one of the most difficult of any that the refrigerating engineer may be called upon to solve. However, it is positive that rough handling, dropping of the core sucking rod into the ice cavity and pounding and pumping of the cans are very bad for the production of sound ice. Sometimes the lowering of the cans quickly into comparatively hot water is better than to do it slowly in cold water, and again, in other cases, the use of 60 deg. F. water in the dip tank and a lowering rate of the can such as would require $7\frac{1}{2}$ minutes to submerge the can has been found best. The old form of sprinkler device at the dump is thought to be more likely to cause checking than the dip tank, whereas the use of air agitation of the dip tank is considered to give better results than the tank without agitation. The accepted reason for can ice cracking is because of freezing stresses due to the expansion and contraction of the water and ice during the freezing process. Some ice is cracked before it is removed from the cans.

Piping.—The method of cooling brine in the ice tank for a long time has been to use direct expansion piping in the brine tank (Fig. 263).

[9a] E. S. Ormsby, National Association of Practical Refrigerating Engineers, 1923.

These pipes are arranged to run the long dimension of the tank, and are placed in stands between the individual ice cans, thereby permitting

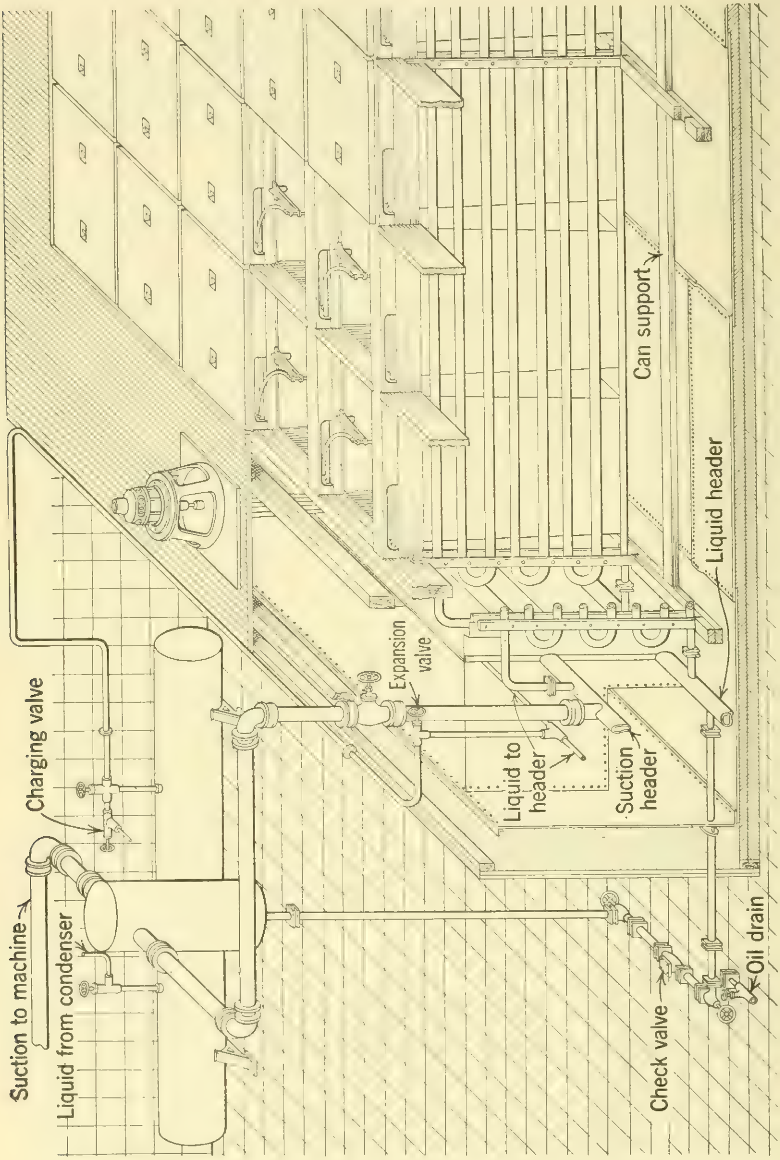


FIG. 263.—Can Ice Making—Tank Assembly.

close proximity between the cans and the heat absorbing medium. These pipe coils (usually $1\frac{1}{4}$-in. pipe) with the old system of operation

were not found to be very effective because they had to be so operated as to prevent liquid returning to the compressor, and the flooded system has entirely superseded it. The flooded system (shown in Fig. 264) is not a device for securing more refrigeration per pound of ammonia but only for providing efficient heat transfer in the evaporating surfaces. Piping which was originally calculated for 12.5 or 15 B.t.u. for the value of the coefficient of heat transfer can transmit twice that amount

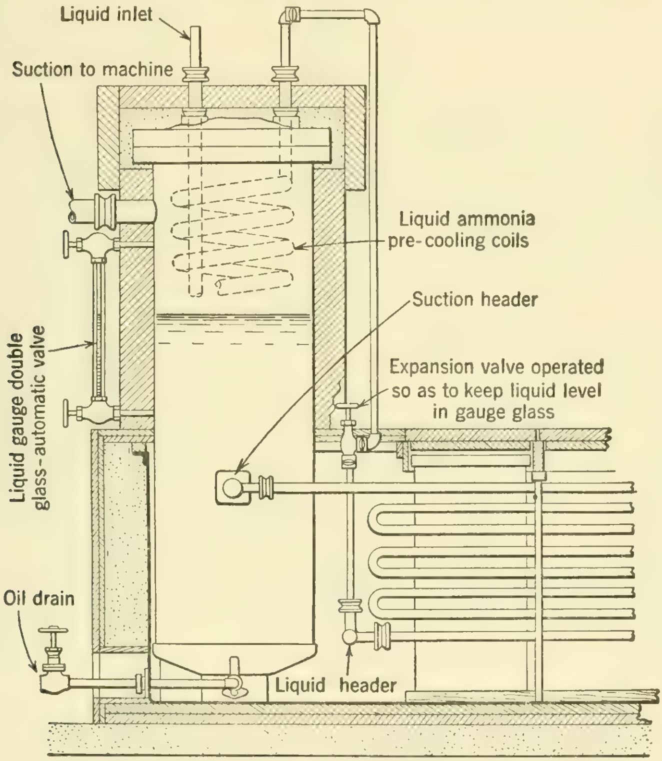


Fig. 264.—The Accumulator and Piping in Ice Tank.

with the flooded system with the same amount of agitation. As piping has been considered to be the cheapest part of the plant it has been the custom to specify 250 to 300 linear ft. of $1\frac{1}{4}$-in. pipe per ton of ice making. For a 100-ton ice plant there would be required under these circumstances from 5 to 6 miles of pipe which must be made very tight unless a loss of ammonia from the system and the formation of ammonia chloride is not to be faced during the ice making season without any means of repair except a complete shut-down and the removal of the

brine from the tank. In the old ice tank designs the process of filling and replacing the cans was very likely to injure the piping.

The alternative arrangement, which is becoming popular in certain sections, is to use the shell and tube brine cooler, submerged in the brine, and arranged for a single pass of the brine. The ammonia, instead of boiling out of a 1500-ft. length of pipe, has a distance of 8 to 10 ft. to travel from the expansion valve to the beginning of the suction return line to the compressor (Fig. 171). The objections to the use of the shell and tube brine cooler have been a fear that the cooler could not be made tight, especially where the tubes were expanded into the tube sheet, and that as the cooling surfaces were only at one end (or both ends) of the tank, the brine temperature would be non-uniform in the tank. There is also a feeling that the resistance to the flow of the brine through the cooler is too great, requiring thereby too much expenditure of power to provide circulation of the brine around the tank and the cans. Apparently some of these fears are not justified as a brine temperature variation of not more than $\frac{1}{2}$ deg. F. is usually found if good agitation is provided, and at the present time there seems to be no difficulty in being able to make both brine coolers and condensers of the shell and tube construction and to keep the tubes tight in the tube sheet by the simple expedient of expanding into the sheet. The shell and tube cooler for this work is shown in Fig. 172, which is designed for a single pass. The usual velocity of the brine in the tubes is 400 ft. per min., and the coefficient of heat transfer under these conditions is 90 B.t.u. per hour. The objection that valuable space will be taken up by the cooler is answered in a way by the statement that as the piping

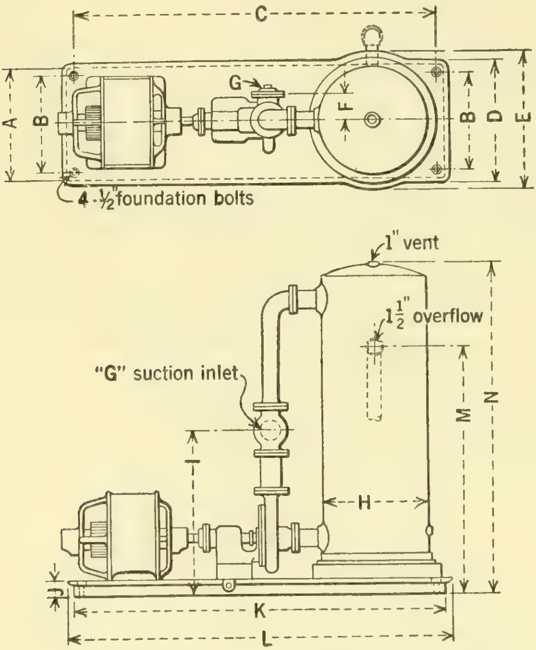


Fig. 265.—Core Sucker.

is not placed between the cans, these can be placed on closer centers. This is especially true in the case of cans in baskets which permit a lighter tank frame construction. The shell brine cooler lends itself to the flooded operation, and by means of the separator and drain (Fig. 170) back to the cooler the danger due to a " slop over " to the compressor is removed.

Precooling Water for the Cans.—If the water goes to the cans at 70 deg. F., then 38 B.t.u. must be removed per pound before freezing can begin, and this action of cooling can be performed more advantageously in some other device, as, for example, in a special water

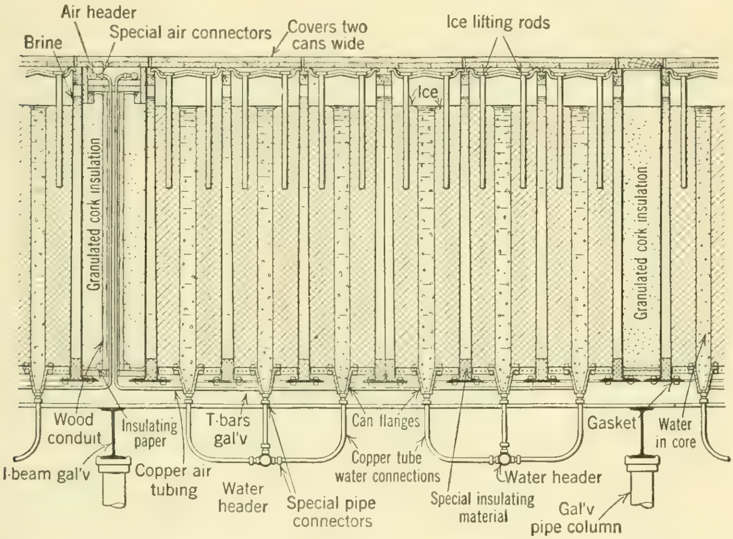


FIG. 266.—The Arctic-Pownall System.

cooler. It should be kept in mind, however, that the use of a water cooler may not give good results unless operating care is observed. It is easy to lose all the advantage of precooling. Tests indicate that the water in the cans, if these are filled at 70 deg. F., can be lowered to 32 degrees in about 15 to 20 minutes. In the Arctic-Pownall process (Fig. 266) the precooling tank is carefully insulated and only slight loss will be occasioned on account of heat leakage from the air to the water.

Freezing Processes.—A number of designs used in the manufacture of ice a few years ago are spoken of but rarely now. The *Hume* froze a cake of four times the usual size (22 in. by 44 in.) which was sawed into standard sized cakes so as best to eliminate the core. Core sucking and the filling of the cavity with distilled water was spoken of as the

Ulrich process. The *Bishop* process, a modification of the plate in order to fit the conditions of the can system, was arranged so that the water entered the can at the bottom and overflowed at the top. As the can was kept cool in the usual manner with brine, freezing took place from all four sides and a fairly clear cake of ice resulted. The can was arranged so as to be stationary and the ice had to be loosened in the can by circulating warm brine and the cakes lifted by means of hooks frozen into the ice for that purpose. The *Quick-freeze* system attempted to cut down the time of freezing by the use of a special pipe arrangement

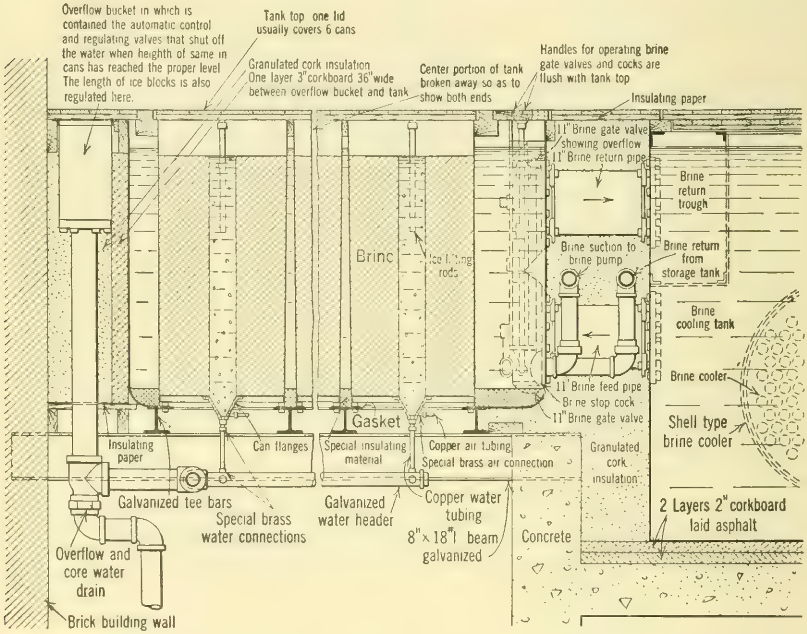


FIG. 267.—The Arctic-Pownall System.

projecting upward from the bottom of the can and designed to circulate brine. As heat was absorbed by each pipe as well as by the usual process, the time of freezing of a standard cake of ice would be reduced. In the plate system there was the *Hill-Ray* method, the only apparent advantage of which was an arrangement of wedges by which the size of the cake was reduced and the amount of sawing diminished in proportion. The *Center-Freeze* method froze the cake from two sides at the same time, the pipes being spaced about 5 to 6 in. apart, and it was claimed that a reduced time of freezing was accomplished. The finished cake would of course have holes in it where the pipes had been during freezing.

The *Holden* Regulation system is still spoken of as a present-day possibility. In this a hollow drum, kept cold by the use of brine, or direct expansion, is made to revolve with its surface in contact with a trough of water (see Lard Cooling, Chapter XIV). The ice film is scraped off and the snow is compressed hydraulically to the size of cake desired. This method is very nearly ideal, the cost per ton is low, and the process is continuous. Unfortunately there is no known means of producing thereby a crystal cake, and this method has not been used commercially for years. Finally, there is the *Jewell* process, a modification of which (the Arctic-Pownall) is a most successful present-day design.

The Jewell process uses a double-walled can which is stationary. The space between the two walls is filled with brine, the brine entering at the top and leaving at the bottom. The bottom of the can is insulated and contains the air connections for agitation. The Arctic-Pownall arrangement is shown in Figs. 266 to 268. The brine is cooled by means of shell and tube brine coolers, and the agitation is vigorous. The ice tank needs to be open at the bottom, and is supported on I beams, as shown. The filling of the cans, dropping, and the refilling of the cores are performed by the means of piping connected to the can flanges. A number of 300- or 400-lb. cans—from 28 to 84—are connected so as to form an ice tank. These cans are all interconnected so that can filling, core dropping, etc., can take place in unison by the operation of one valve. By this method all cans are filled at the same level which can be adjusted, and the water for can filling can be stored in a tank to the correct amount (previously cooled if desired).

The usual method of operation is as follows: The plant is made up of a number of tanks all of similar construction. The water for the cans of one of these tanks has been precooled by an arrangement shown in Fig. 268, and by the operation of one valve the tank is emptied and the cans in one of the tanks are filled. The air agitation is started in all the cans of the tank by the opening of one valve and the brine is circulated about the cans by the operation of two quick-acting valves. The advantages of this system are in the much reduced labor costs, as one man is claimed to be able to pull at the rate of 175 tons per 24 hours, no mechanical injury to the cans, no brine on the covers of the tank, economy in the use of brine in the precooling of the water for the cans and by the use of this heated brine for the thawing of the ice out of the cans, the use of low-pressure air for agitation entirely and no chance of white (marble) ice due to shut-downs of power. The first cost is considerably greater (10 to 15 per cent) than the usual medium-pressure air can system, and the design calls for vigorous brine agitation. However, it is possible to operate with less loss of refrigeration, with less wear

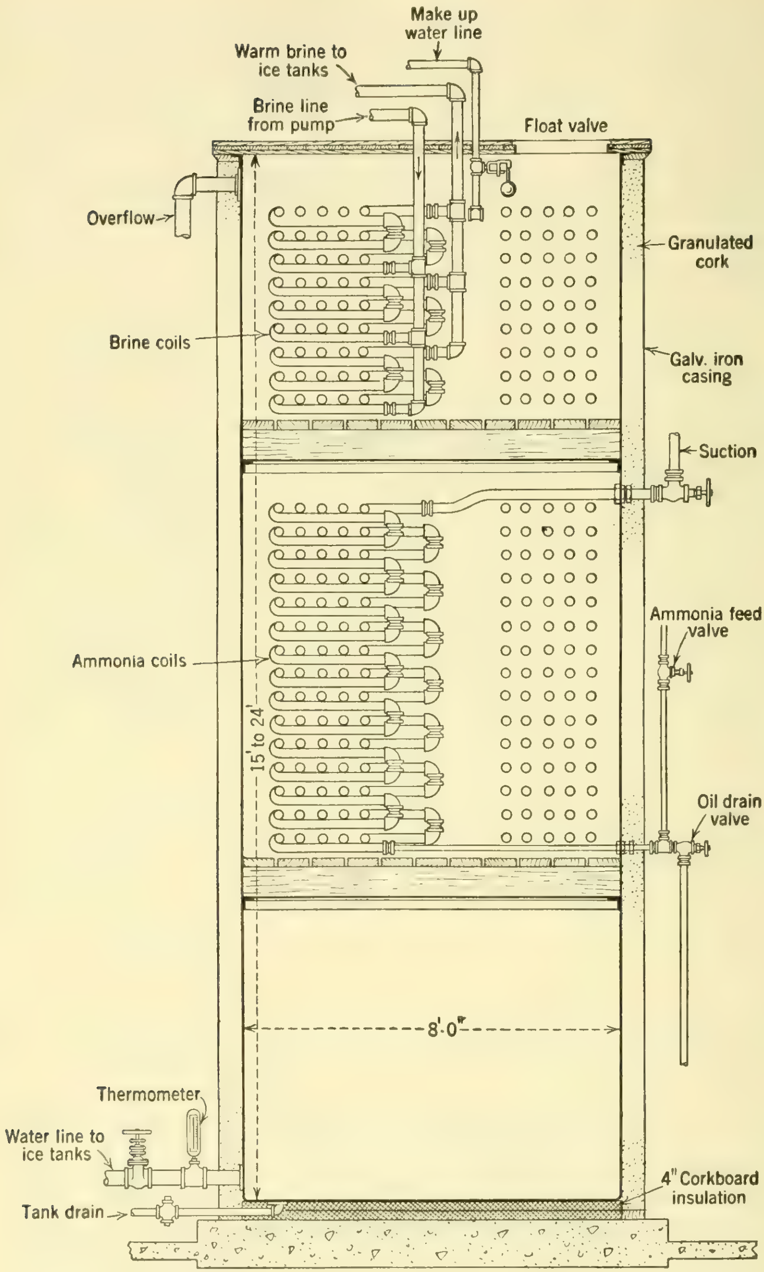


FIG. 268.—Water Cooler and Tank.

and tear on the equipment, and with less labor on the tank room floor with this system than with any other.

The Design of an Ice Making Plant.—Ice plant design is different in various parts of the United States and even in different parts of the same city. In order that design details may be emphasized, some material already treated in this chapter will be repeated as far as it affects the matter of design.

TABLE 91

FREEZING TANK FRAMEWORK AND COVERS FOR STANDARD CANS

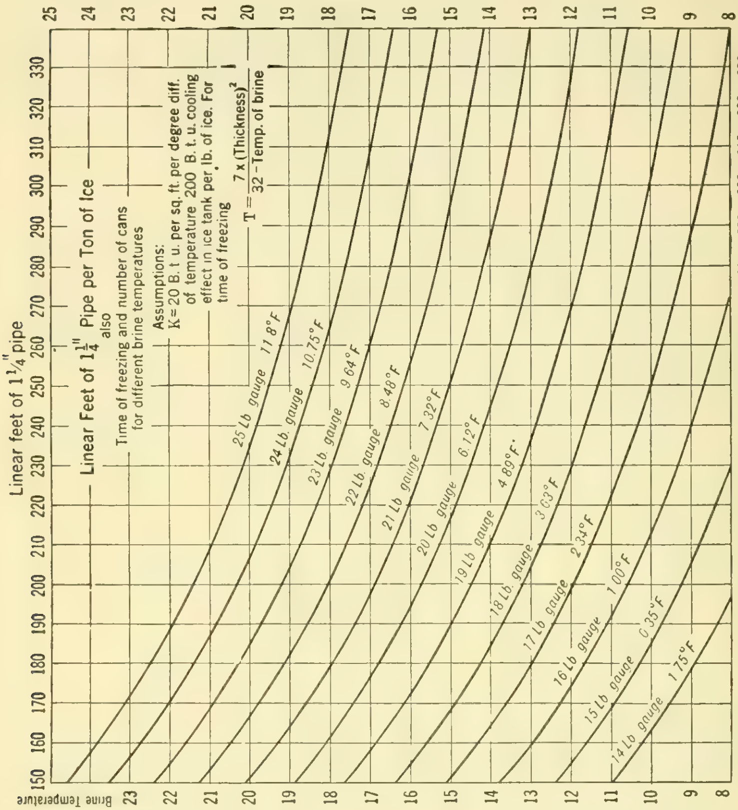
Weight of Cake of Ice, Pounds	Can Dimensions Measured at Top, Inches		Framework Dimensions, Inches, Center to Center		Covers, Weight per Can, Pounds	
	Width	Length	Width	Length	Distilled or opaque	Raw water
25	4	9	7	11	28	28
50	5	12	8	14	30	30
60	5	14	8	16	30	30
100	8	16	11	18	32	32
200	11½	22½	14½	24½	36	36
300	11½	22½	14½	24½	36	36
400	11½	22½	14½	24½	36	36

NOTE.—Covers and frames are made of finished lumber. Covers are made of two thicknesses of 1¼-in. material screwed together with wood screws and have recessed opening with pin for lifting hook. Unless otherwise specified material will be oak for all standard equipment.

The present tendency is towards the reduction of the cost of labor to the minimum. This means the use in the larger plants of the 400-lb. cans instead of the 300-lb. cans, as the time of freezing is the same for both, and—with the exception of certain patented tank designs—of the use of medium or high-pressure air agitation. In fact, it has been pointed out that certain raw waters heavy in certain salts cannot make the best ice with low-pressure air agitation.

The tendency of the times is also for the lifting of gangs of ice cans at a time, say from five to eighteen or more, though the general practice is not more than six. The advantage of lifting more than two cans at a time depends on local conditions, and the entire subject is one for careful analsysis if the most economical arrangement is to be made. The heavy

agitation is for the purpose of increasing the coefficient of heat transfer of the pipes in the tank from 20 B.t.u. (for flooded operation) per square



Brine Temperature	50 Pound Cans: Time to freeze	50: No cans per ton	100: Time to freeze	100: No cans per ton	200: Time to freeze	200: No cans per ton	300: Time to freeze	300: No cans per ton	400: Time to freeze	400: No cans per ton
23	25.7	42.9	45.2	37.7	94.2	39.2	94.2	26.2	94.2	19.6
22	23.1	38.5	40.7	34.0	84.7	35.3	84.7	23.5	84.7	17.7
21	21.0	35.0	37.0	30.8	77.0	32.1	77.0	21.4	77.0	16.1
20	19.3	32.2	33.9	28.3	70.6	29.4	70.6	19.6	70.6	14.7
19	17.8	29.7	31.3	26.1	65.2	27.2	65.2	18.1	65.2	13.6
18	16.5	27.5	29.0	24.2	60.5	25.2	60.5	16.8	60.5	12.6
17	15.4	25.7	27.1	22.6	56.5	23.5	56.5	15.7	56.5	11.8
16	14.5	24.2	25.4	21.2	52.9	22.1	52.9	14.7	52.9	11.0
15	13.6	22.7	23.9	19.9	49.8	20.8	49.8	13.8	49.8	10.4
14	12.9	21.5	22.6	18.9	47.1	19.6	47.1	13.1	47.1	9.8
13	12.2	20.3	21.4	17.9	44.6	18.6	44.6	12.4	44.6	9.3
12	11.6	19.3	20.4	17.0	42.3	17.6	42.3	11.8	42.3	8.8
11	11.0	18.3	19.4	16.2	40.3	16.8	40.3	11.2	40.3	8.4
10	10.5	17.5	18.5	15.4	38.5	16.0	38.5	10.7	38.5	8.0
9	10.1	16.8	17.7	14.8	36.8	15.3	36.8	10.2	36.8	7.7
8	9.6	16.0	17.0	14.2	35.3	14.7	35.3	9.8	35.3	7.4

Fig. 269.—Ice Freezing Diagram.

foot per one degree difference per hour (the value k) to 25 or possibly to 30. The particular gain here is to permit the amount of piping in the tank to be reduced, and also to reduce the time of freezing of the water

TABLE 92

Simple Belted Air Compressors for Medium-Pressure Raw Water Plants

Number of Cans	Tons of Ice	Cubic Feet of Air per Minute	Size, Inches		R.p.m. Required	Horsepower Motor Required	Maximum Compressed Speed, R.p.m.	Maximum Pressure for Comp., Pounds	Pulley, Inches		Pipe Connections, Inches			Approximate Floor Space, Inches	Weight Boxed
			Diameter	Stroke					Diameter	Face	Suction	Discharge	Water		
70	5	14.0	4	5	250	3	350	125	30	$4\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	55×22	850
84	6	16.8	5	5	225	3	350	125	30	$4\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	55×22	850
112	8	22.4	6	5	200	5	350	100	30	$4\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$\frac{1}{2}$	57×22	950
144	10	28.8	7	5	205	5	350	50	30	$4\frac{1}{2}$	2	2	$\frac{1}{2}$	59×22	1060
168	12	33.6	7	5	215	5	350	50	30	$4\frac{1}{2}$	2	2	$\frac{1}{2}$	59×22	1060
210	15	42.0	7	5	225	$7\frac{1}{2}$	350	50	30	$4\frac{1}{2}$	2	2	$\frac{1}{2}$	59×22	1060
280	20	56.0	7	6	240	10	350	100	36	$5\frac{1}{2}$	2	2	$\frac{1}{2}$	60×26	1305
360	25	72.0	8	6	250	10	350	50	36	$5\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$\frac{1}{2}$	68×26	1475
420	30	84.0	9	6	230	15	325	25	36	$5\frac{1}{2}$	3	3	$\frac{3}{4}$	70×26	1550
484	35	96.8	9	8	230	15	325	100	42	$8\frac{1}{2}$	3	3	$\frac{3}{4}$	81×34	2610
560	40	112.0	10	8	210	20	325	50	42	$8\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{4}$	85×34	2800
720	50	144.0	10	8	235	20	325	50	42	$8\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{4}$	85×34	2800
840	60	168.0	12	8	220	25	325	30	42	$8\frac{1}{2}$	4	4	1	87×34	2925
1120	80	224.0	14	9	200	30	300	25	54	$9\frac{1}{2}$	5	5	$1\frac{1}{4}$	94×39	3800
1440	100	288.0	14	9	210	30	300	25	54	$9\frac{1}{2}$	5	5	$1\frac{1}{4}$	94×39	3800
1680	120	336.0	16	10	200	40	275	35	60	$10\frac{1}{2}$	6	6	$1\frac{1}{4}$	107×46	6000
2160	150	432.0	16	10	215	50	275	35	60	$10\frac{1}{2}$	6	6	$1\frac{1}{4}$	107×46	6000
2880	200	576.0	14	9	210	30	300	25	54	$9\frac{1}{2}$	5	2	$1\frac{1}{4}$	94×39	7600
			14	9	210	30	300	25	54	$9\frac{1}{2}$	5	5	$1\frac{1}{4}$	94×39	

Based on 14 cans per ton. For less figure .2 cu. ft. air per minute and use proper compressor, working pressure of compressors 25 lb. Based on belted, enclosed type, horizontal, D. A. compressor with unloader and foundation bolts. No mechanically operated valves included.

in the cans, although the empirical formula for the time of freezing has no term to denote the effect of agitation on the freezing time. The latest practice is to raise the evaporating or suction pressure as high as possible, say, from 22 to 25 lb. gage. This means a high boiling temperature of the ammonia and in consequence greater capacity for the compressor and less cost in horse power per ton of refrigeration, but this of course increases the auxiliary horse power. The relation of number of cans, brine temperature and pipe surface is well shown in Fig. 269 and Table 92.

The matter of the calculation of an ice plant problem may, then, be different in certain details, but the manner of the calculation can be shown clearly by the problem, the details of which would need to be carefully decided for the individual case by an analysis of the costs, by a consideration of the power and labor items, as follows: The ice plant is to be a 100-ton raw water can ice plant, for 24-lb. suction and 154.5-lb. condenser pressure, using 70 deg. F. water to the cans and 16 deg. brine. The outside air temperature is 90 degrees, and the temperature of the earth is 50 deg. F. Use high pressure air agitation, 30 lb. at the compressor and 15 lb. at the cans. Two dehumidifiers, brine cooled, to cool 200 cu. ft. of free air from 90 to 20 deg. F. per minute.

Two tanks[10] of $\frac{1}{4}$-in. steel will be used, 5 ft. 1 in. high, each to hold 600 400-lb. cans 58 in. high. Each tank will have the one accumulator and two 12-in. brine agitators—using 5 hp. each—with motors direct connected to vertical shafts. Can centers are to be 2 ft. $\frac{3}{8}$ in. by $14\frac{1}{4}$ in., and the tanks are to be insulated with 12 in. of granulated cork on the sides and with 5 in. of cork board on the bottom. Lift five cans at a time, and use $1\frac{1}{4}$-in. full weight pipe.

1st. To Find the Heat Removed per Pound of Ice Placed in Daily Storage.

To cool the water from 70 to 32 deg. F. per 1 lb. of ice....	38.0	B.t.u.
To freeze the water per 1 lb. of ice....................	144.0	"
To cool the ice from 32 to 16 deg. (spec. heat of ice = 0.5)	8.0	"
Total........	190.0	B.t.u.
Add 10 per cent for non-computable losses.............	19.0	

Heat leakage through the insulation (calculated as in the problem on cold storage)[11]
Sides, 12-in granulated cork, 2-$\frac{7}{8}$ in. and g. cypress.

$$U = \frac{1}{\frac{1}{1.4} + \frac{2}{1} + \frac{12}{0.296}} = 0.0231.$$

[10] These details follow standard practice. [11] See Chapter XIV.

Bottom, 5-in. corkboard, 9-in. concrete.

$$U = \frac{1}{\frac{5}{0.308} + \frac{9}{5.3}} = 0.0558.$$

Heat leakage: Sides $1187 \times 0.0231 \times (90 - 16) = 2030$ B.t.u.
Bottom $3420 \times 0.0558 \times (50 - 16) = 6490$ "

Total = 8520 B.t.u.

$$= 4260 \text{ B.t.u. per hour per tank of } = \frac{4260 \times 24}{2000 \times 50} = 1.02 \text{ B.t.u. per lb. of ice.}$$

The air for agitation is cooled after compression first by water and then by brine (using two tanks with a four-way valve for alternate change of direction of flow). The brine in refrigerating the air adds a certain amount to the load on the compressor, as, for example, the heat removed from each pound of the air may be considered as 35.0 B.t.u., and the specific volume of the air as 14.2 cu. ft. Then

$$\frac{200}{14.2} \times 35.0 = 493 \text{ B.t.u. per minute,}$$

or, in terms of the ice made,

$$\frac{493 \times 60 \times 24}{100 \times 2000} = 3.56 \text{ B.t.u. per lb.}$$

The agitation of the brine, as already stated, assists in the heat transfer, and keeps the brine temperature constant. Agitation always tends to heat up the brine, and it is usual to figure that the entire heat equivalent of the pump horse power is converted into heat and gas to be neutralized (refrigerated) by the action of the compressor. In this case the amount becomes

$$\frac{20 \times 42.4 \times 60 \times 24}{100 \times 2000} = 6.1 \text{ B.t.u. per lb. ice.}$$

The total refrigeration per one pound of ice made is then

$$209 + 1.02 + 3.56 + 6.1 = 219.7 \text{ B.t.u.}$$

In round numbers the value 220/144 = 1.53 is the ratio of the number of tons of refrigeration capacity required per ton of ice made according to the calculation. The value generally used is 1.6, although some engineers go to the extreme by the use of the ratio 2.0, but such a

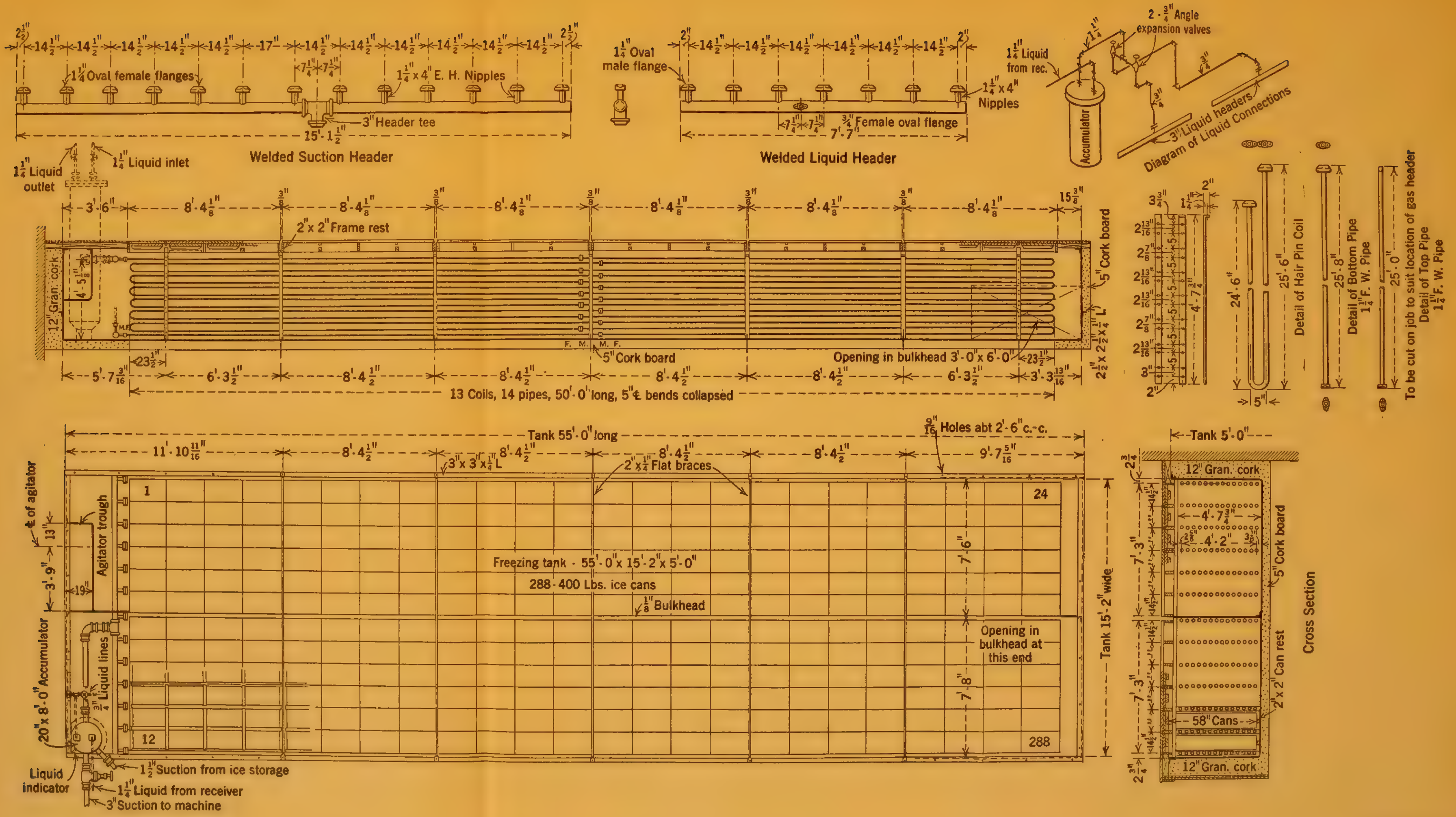


FIG. 270.—Can Ice Tank Design.

To face page 416.

to consider first the mechanical efficiency of the equipment; second, the cost of the investment; and third, the plant operation. Medium-sized plants have produced ice at the rate of 37 to 38 kw.-hrs. per ton of ice as the yearly average with a load factor of 60 to 70 per cent, but the cost of power is only one of a number of costs. In all likelihood it is possible that with good agitation and with the flooded system (which makes it possible to decrease the amount of gas in the evaporating system, and by having a heavy feed with some liquid in the coils for the entire length of the direct expansion piping) a value of 30 or more can be obtained for k,[12] but a design of this sort requires more care in the operation than does the ordinary plant.

Selecting the Compressor and the Compressor Drive.[13]—Selecting the compressor and the compressor drive has become a difficult problem in these days, with the many types and designs which prevail. However, there are a few points of particular prominence which are worth mentioning. It is a mistake to consider that clearance in an ammonia cylinder means an appreciable loss of economy. Clearance affects the capacity of a compressor, but it has not been proved that it increases the horse power per ton of refrigeration to any great extent except for the friction of the machine which remains nearly constant though it is *relatively* larger at small than at full loads, per ton of refrigeration.

The space allotted to the compressor frequently decides the type of compressor. The principle of the design makes one feel that the vertical single-acting compressor using the suction valve in the piston is the correct design in order to reduce the superheating of the suction gas by the cylinder walls to a minimum, but this is also true of the horizontal double-single-acting design. In each case there is a uniflow principle of gas flow. The factor 0.86 for the volumetric efficiency is a term to allow for this, although the figure mentioned may be a little low for the vertical single-acting compressor. However, one is led to believe that the semi-enclosed type of vertical compressor is becoming much less popular than, say, 10 years ago. It has a reciprocating rod type of stuffing-box which is much more difficult to keep tight than the rotary type used in the enclosed compressors and is likely to have more friction in the stuffing-box. In Great Britain the semi-enclosed type is not used at all, and the modern type of horizontal compressor has not been built until recently, while the three- and four-cylinder enclosed machine is very popular and uses forced lubrication throughout. In the United States every manufacturer has a design of a small compressor of the vertical

[12] Thomas Shipley, National Association of Practical Refrigerating Engineers, 1926.

[13] See Chapters II and XIX.

single-acting type, but the majority have horizontal compressors only for the larger sizes. The York Manufacturing Company has developed an enclosed compressor of large capacities, using three cylinders, and going up from 300 to 400 tons of refrigeration capacity.

As regards the drive, for electric-driven compressors[14] requiring 75 hp. and over, a synchronous motor generally should be used with direct connection to the shaft. The gain is in a greater overall efficiency as there are no belt or other transmission losses which in other cases usually amount to from 6 to 8 per cent. This type of motor is particularly good because of its ability to improve the power factor of the circuit as the motor may be made to operate as a synchronous condenser. The disadvantage, which is true also of the squirrel-cage induction and other types of motors, is the impossibility of varying the speed of the compressor. Variation of the capacity can then be obtained only by the judicious use of clearance pockets, a device which has become quite usual. If a steam engine is used, the uniflow engine is the most popular because of the flat water rate curve and the low steam consumption per horse power hour. If the oil engine is decided on there are a number of types to choose from. In any case the power delivered to the shaft should be (see Fig. 31 for 24-lb. suction and 154.5-lb. condenser pressures) 1.10 hp. per ton of refrigeration.[14a] Each of the two compressors will have a capacity of $76.6 \times 1.1 = 84.3$ hp. at the shaft. The piston displacement may be found from the same set

[14] See Chapter XXIII.

[14a] The capacity and the horse power per ton (of refrigeration) can be calculated as follows, for 24.0 lb. gage suction and 154.5 lb. gage condenser pressures:

$$i''_{24\text{ lb. g.}} - i'_{86\text{ deg. F.}} = 614.9 - 138.9 = 476 \text{ B.t.u.}$$

$$= \text{net refrigeration per 1 lb. liquid ammonia.}$$

Theoretical volume boiled off per ton of refrigeration per minute $= \frac{200}{476.0} \times 7.28$ $= 3.06$ cu. ft., and with a volumetric efficiency of 0.86 the piston displacement per ton per minute becomes $3.06/0.86 = 3.56$ cu. ft., not allowing for clearance. (See Chapter II.)

The horse power per ton of refrigeration is obtained from

$$W = \frac{n}{n-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$= \frac{1.28}{0.28} \times 144 \times 38.7 \times 3.56 \left[1 - \left(\frac{169.2}{38.7}\right)^{\frac{0.28}{1.28}}\right]$$

$$= 34{,}500 \text{ ft.-lb.} = 1.05 \text{ hp.}$$

of curves, and for the conditions stated this becomes 3.6 cu. ft. per ton of refrigeration per minute. Therefore, a 76.6-ton machine will require $76.6 \times 3.6 = 276$ cu. ft. per min. for each of the compressors. The type of the compressor chosen will affect the rotative speed, but the present practice is to use some form of light weight valve of the plate or the ribbon type. The heavy poppet valve is practically obsolete. Should a synchronous motor be used clearance pockets may be employed to advantage in regulating the capacity of the compressor.

Ice-making plants have use for *two suction pressures*,[15] the second being on the fore-cooler at, say, 35-lb. gage. This is especially true

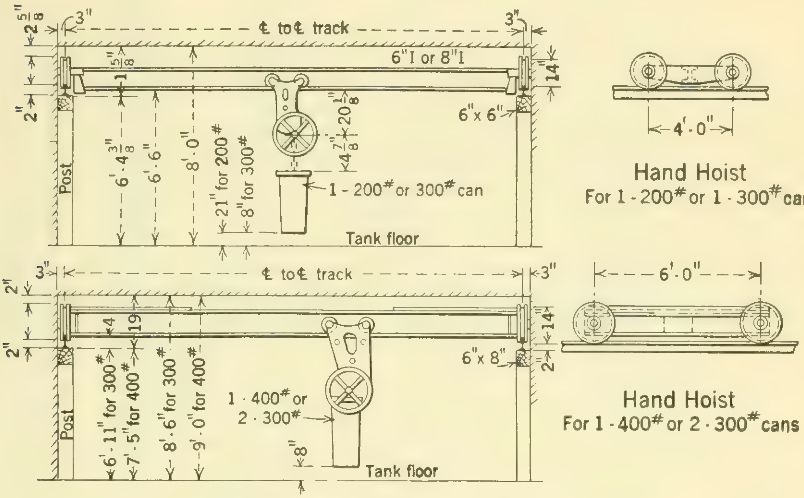


FIG. 271.—Hand-Operated Can Hoist.

where the water for the cans is 70 deg. F. or higher and where suitable care is taken to prevent much loss due to the warming of this water between the fore-cooler and the cans. If the operating conditions were constant the multiple-effect compressor could be used to advantage here, but the conditions are not constant for any length of time. The result is that either a separate machine is used for precooling the water for the cans, or the attempt is made—as in the design above—to increase the suction pressure to as high a pressure as possible.

In choosing the details of the air agitation and harvesting, the costs[16] should be carefully considered, as well as other local conditions. The high-pressure air system was selected for the problem, and also multiple

[15] See Chapter II.

[16] See first part of the chapter.

can lifting and filling devices. These matters should be given careful consideration. Figure 271 gives details for the hand-operated hoist. At

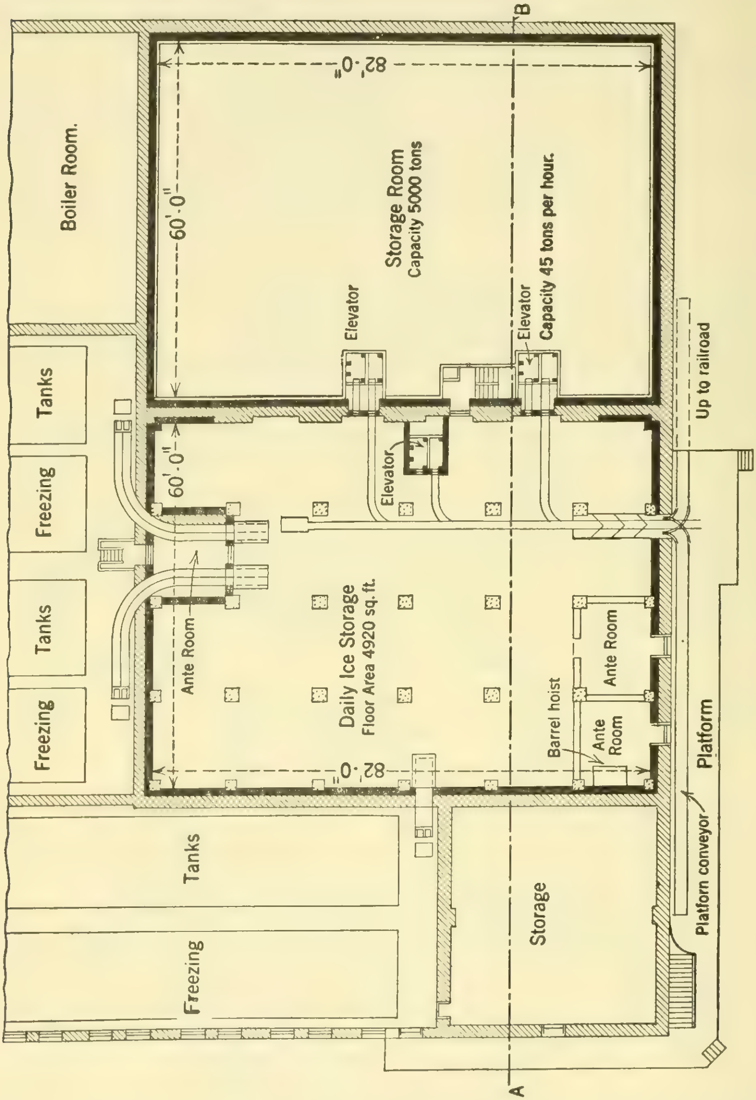


FIG. 272*a*.—Typical Modern Ice Plants.

the present time this type would be used only for the smaller plants. The larger plants would use the *air* or the electric motor-driven hoist.

Ice-making Plants.—Figures 272, 273 and 274 give details of modern ice-making and ice-storage plants. All plants have a *daily* ice storage,

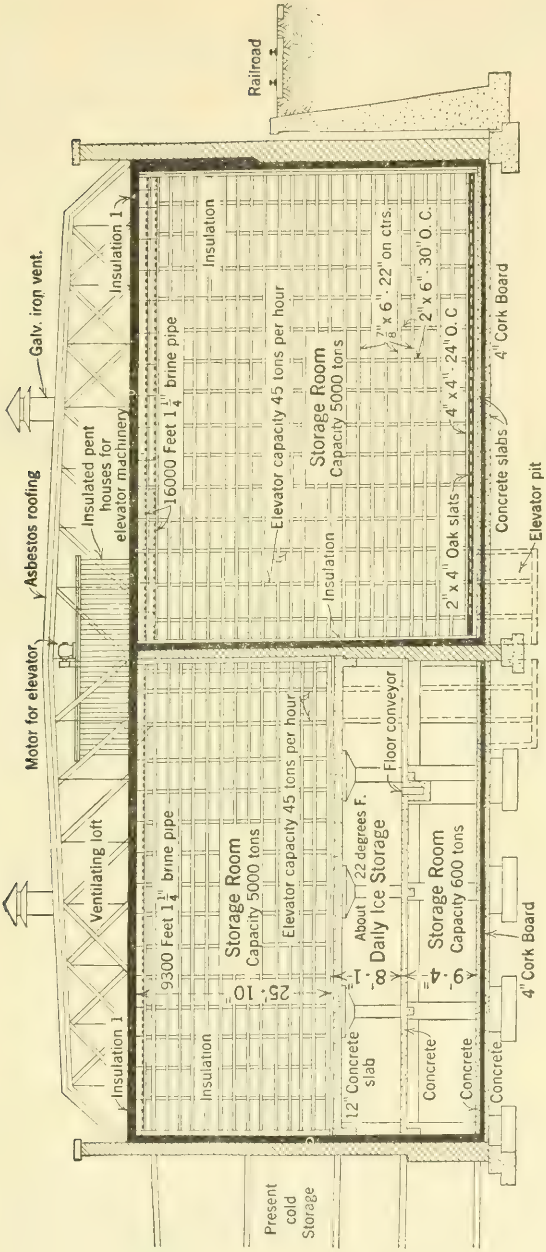


FIG. 272b.—Typical Modern Ice Plants.

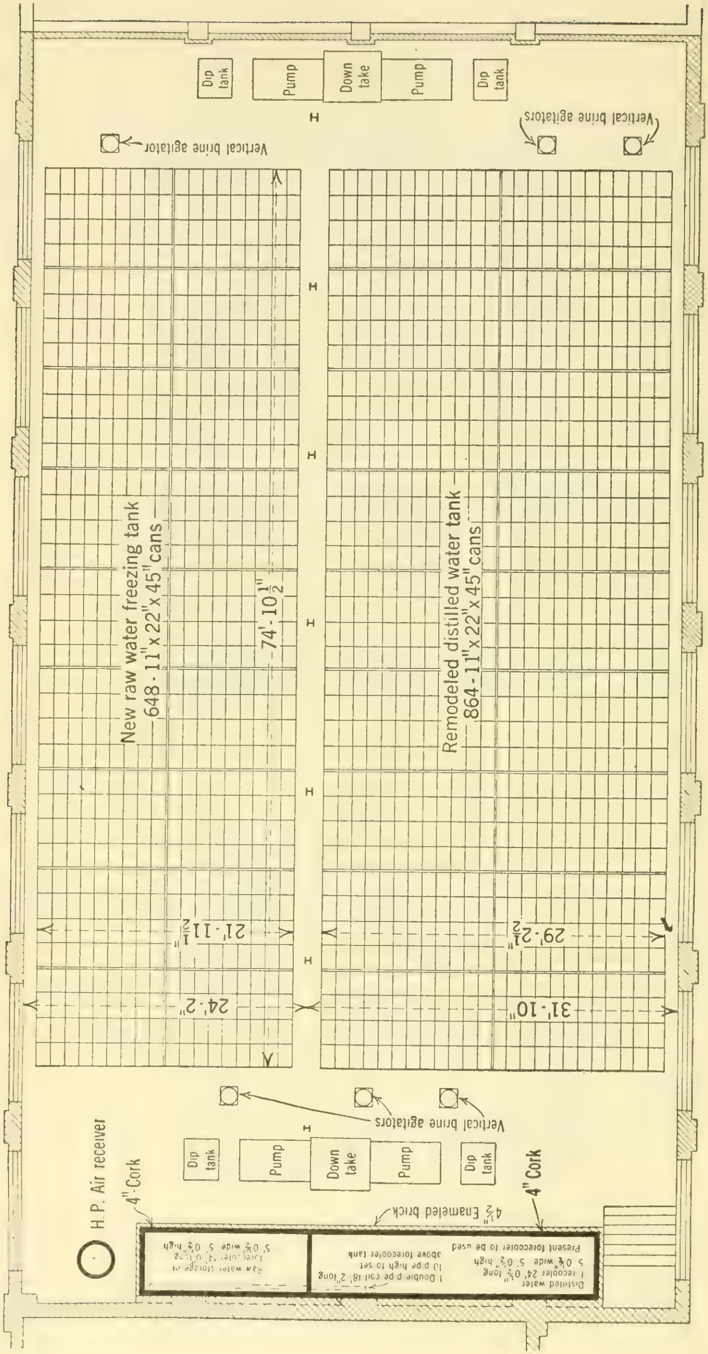


FIG. 273a.—Typical Modern Ice Plants.

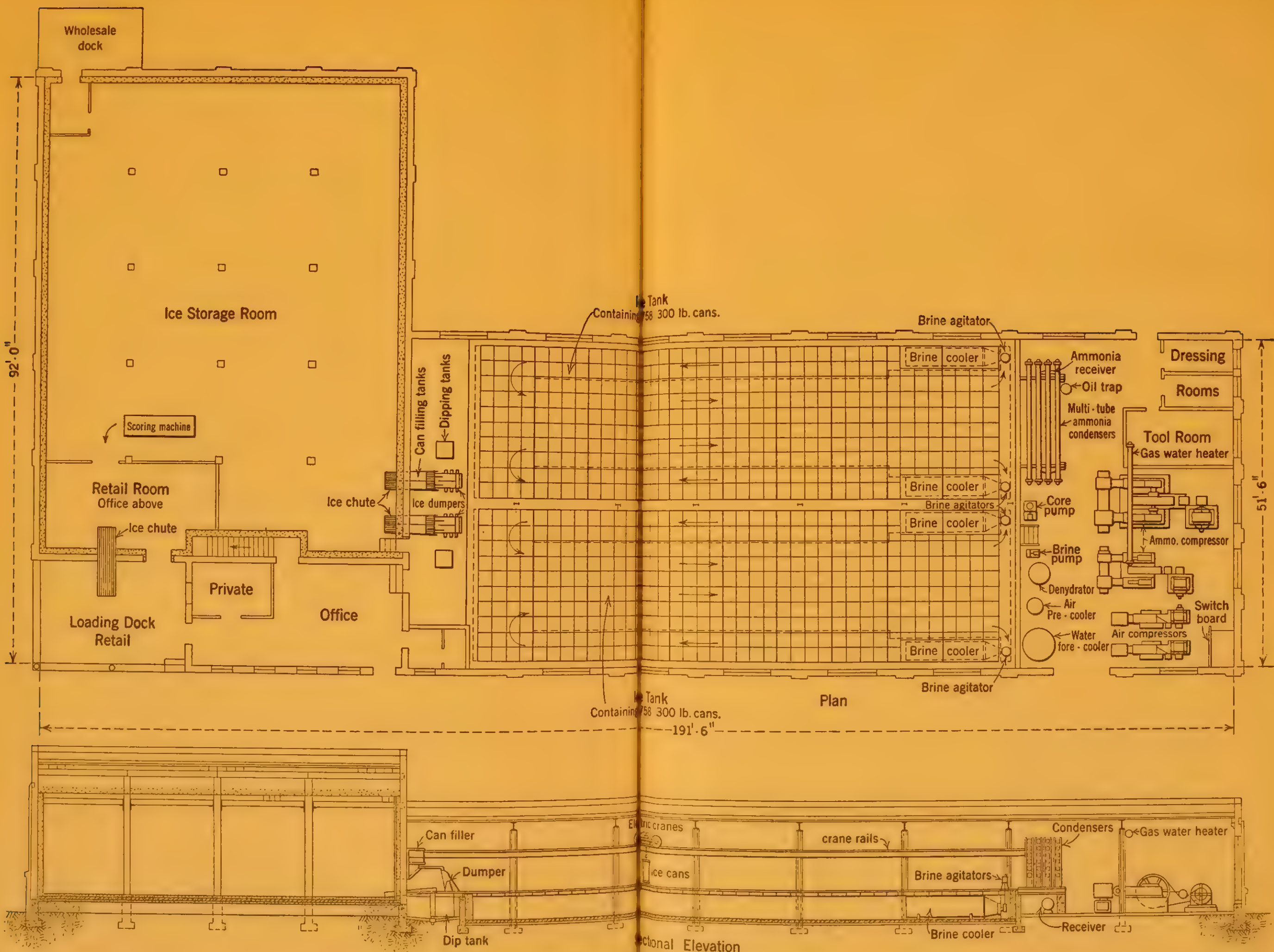


Fig. 274.—Typical Modern Ice Plants.

To face page 423.

and most of them have a storage sufficient in capacity to permit an emergency shut-down of from three to four days. It is a question whether the large ice storage warehouse, using a smaller sized compressor for operation from ten to eleven months of the year is more economical than the larger sized compressor and ice tank, to be operated as the demand for ice requires, without the ice storage.

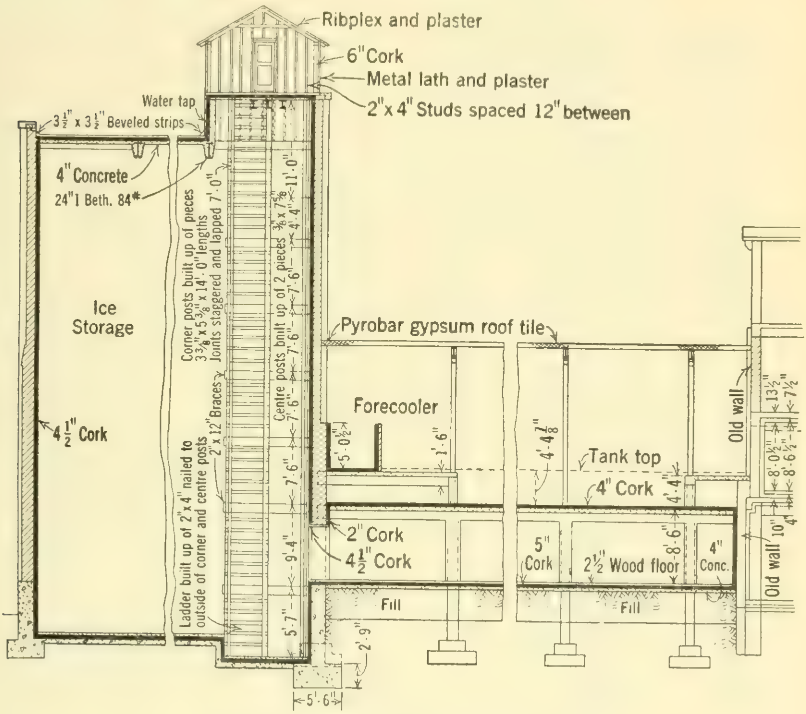


Fig. 273*b*.—Typical Modern Ice Plants.

The following paper, by H. T. Whyte, assistant general manager, engineering and refrigeration, of the Pacific Fruit Express Co., San Francisco, Calif., was presented at the Fifteenth Annual Convention and Exhibition of the N.A.P.R.E. on November 12, 1924.

The capacity of compressors, tanks and storage rooms of plant should be in direct relation to the annual tonnage requirements and, as well, the daily and monthly demand. For instance, where a large daily demand is encountered, which goes on for a period of five months or more, it has been found as a rule more economical to provide for practically the entire requirements through machine capacity, with but small storage room for reserve, in that greater economy can be obtained through pulling requirements direct from tanks, rather than providing large storage room and storing ice throughout the year for disposal during summer months, resulting in excess cost on account of extra handling and, as well, the cost of maintaining

temperatures on storage room throughout the year, and with practically no shut-down period during the winter months.

For example, let us compare the relative merits and costs of operation of a 200-ton daily capacity manufacturing plant, costing approximately $300,000, with a 125-ton daily capacity manufacturing plant, with 10,000 tons storage room, costing $277,000 where the annual demand would be 25,000 tons.

Aside from the saving indicated above (loss account of breakage in storage and

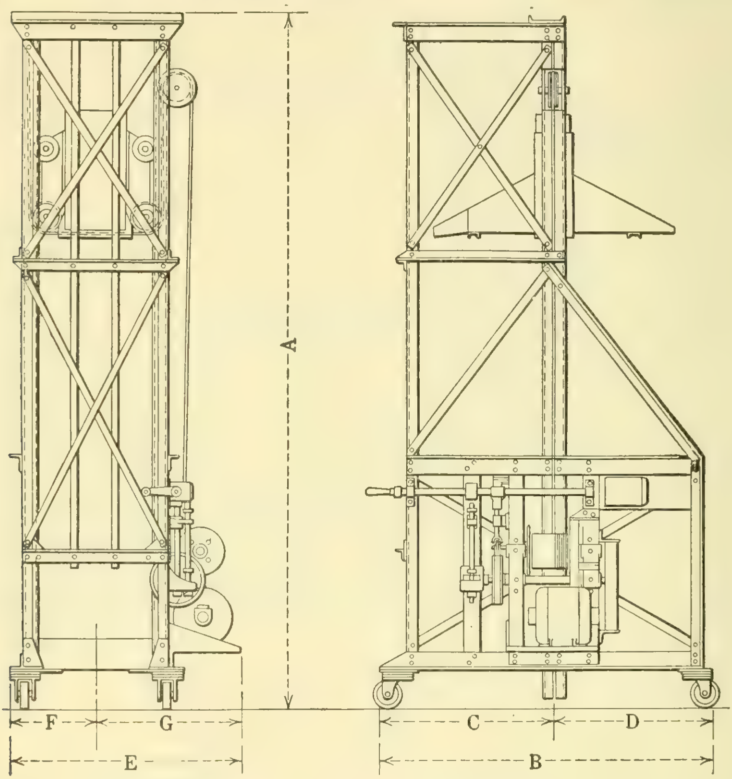


FIG. 275.—The Benching Machine.

removing from storage would be practically eliminated), there is the advantage of supplying your patrons with first quality fresh ice, with nicely squared edges as against stored ice. However, there is the disadvantage of not being able to run continuously even during the peak load period, on account of weather changes which might result in having to shut down part of the plant for a few days, which of course is quite costly.

Climatic conditions will have considerable bearing on the type of installation, with particular reference to type of condensers, and whether cooling tower would prove desirable; also, in the type of construction to be used for tank and compressor room building. In some locations, it is possible to construct this portion of the plant very cheaply, yet serviceably.

Water is one of the most important factors affecting ice manufacturing production costs, for the reason that pure water freezes more rapidly, makes tougher ice, with consequent less breakage and smaller loss due to slush. Also, cold water requires less forecooling for ice making and less is used for condensers; it reduces head pressure with consequent reduction in power consumption. Ample supply of water should also be assured, for lack of sufficient water over condensers is as bad as warm water, in that proper cooling and consequent reduction of condenser pressure cannot be accomplished unless good and sufficient quantity of cold water is available.

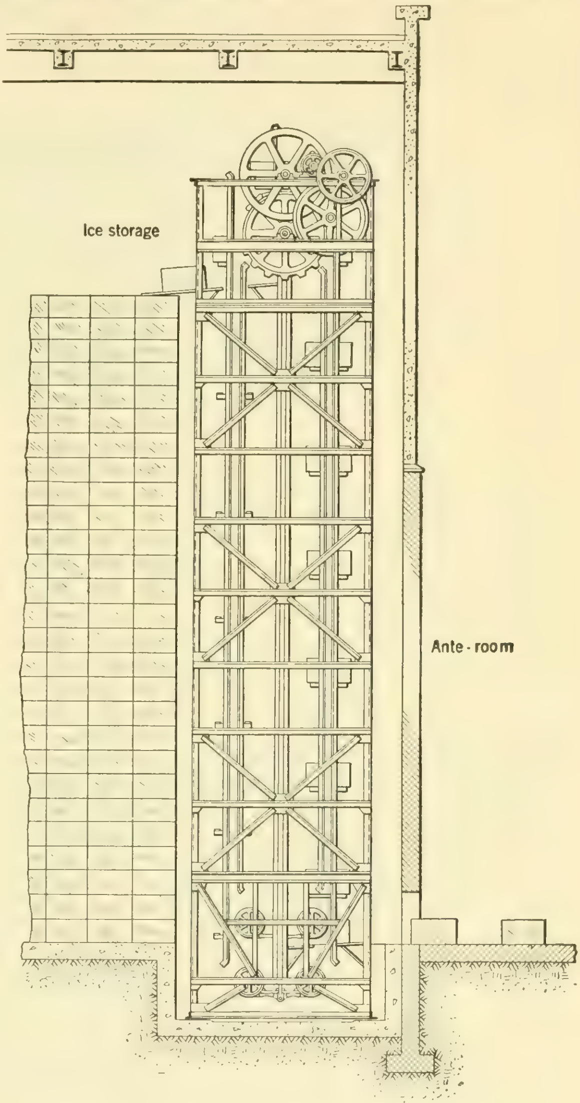


FIG. 276.—Ice Elevating Device.

Ice Storage.—In ice storages two designs of elevating and lowering machines may be used. The daily storage usually has the ice on end, and just one tier. A chain conveyor nearly flush with the floor is used in the daily storage for horizontal moving of the ice. For ice storage rooms of medium height the benching machine (Fig. 275) and for taller storage rooms, as in Fig. 273, a device more like Fig. 276 would be used.

Ice Scoring.—With certain retail trade there is a demand for scored ice. By this is meant that the 300 and 400-lb. ice cakes have had the

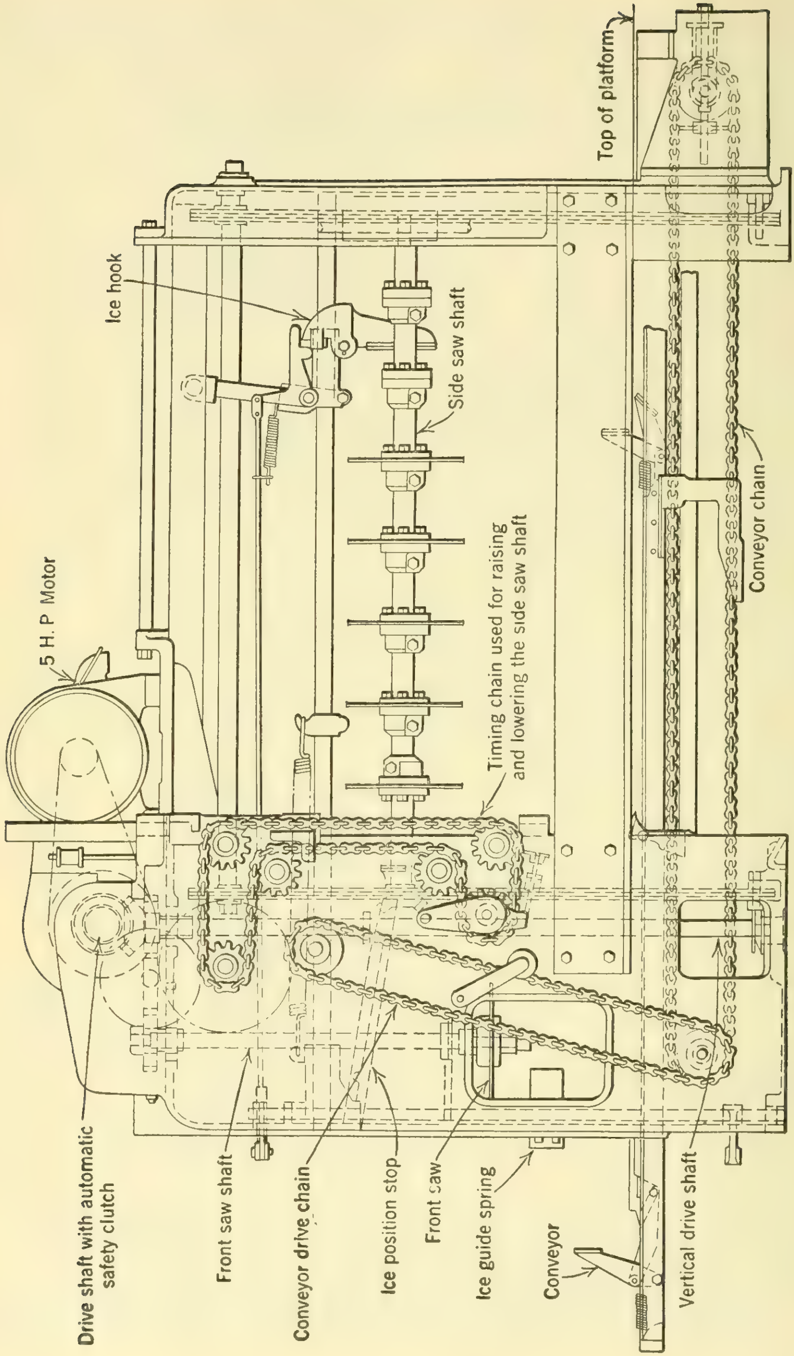


Fig. 277.—Ice Scoring Machine.

proper saw cut so as to register the size of a 25, 50, 75, or 100-lb. piece of ice. Figure 277 shows such a scoring machine.

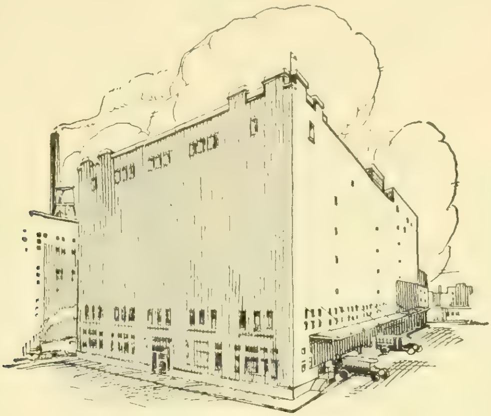


FIG. 278.—Cold Storage Warehouse.

Item	10,000 Tons of Storage			
	200-ton plant		125-ton plant	
	Cost per ton	Total	Cost per ton	Total
General expense, including insurance, taxes, depreciation and interest on investment (14 per cent)........................	$1.68	$42,000	$1.55	$38,780
Plant labor.....................	.10	2,500	.14	3,500
Tank room labor................	.10	2,500	.14	3,500
Day storage room labor..........	.06	1,500	.10	2,500
Power...........................	.45	11,250	.47	11,750
Ammonia........................	.05	1,250	.05	1,250
Water and other supplies.........	.05	1,250	.05	1,250
Maintenance....................	.15	3,750	.15	3,750
Salary of portion of crew during shut-down period.............	.15	3,850*	.088	2,200†
Winter storage room labor, 10,000 tons........................	....	....	.40	4,000
Winter storage room refrigeration, 10,000 tons..................	....	....	.30	3,000
Totals.....................		$69,850		$75,480
Average cost per ton...........	$2.79		$3.02	
Saving in favor of large daily capacity, $0.23 per ton.				

* On basis of 5 months' operation; shutdown 7 months @ $550 per month.
† On basis of 8 months' operation; shutdown 4 months @ $550 per month.

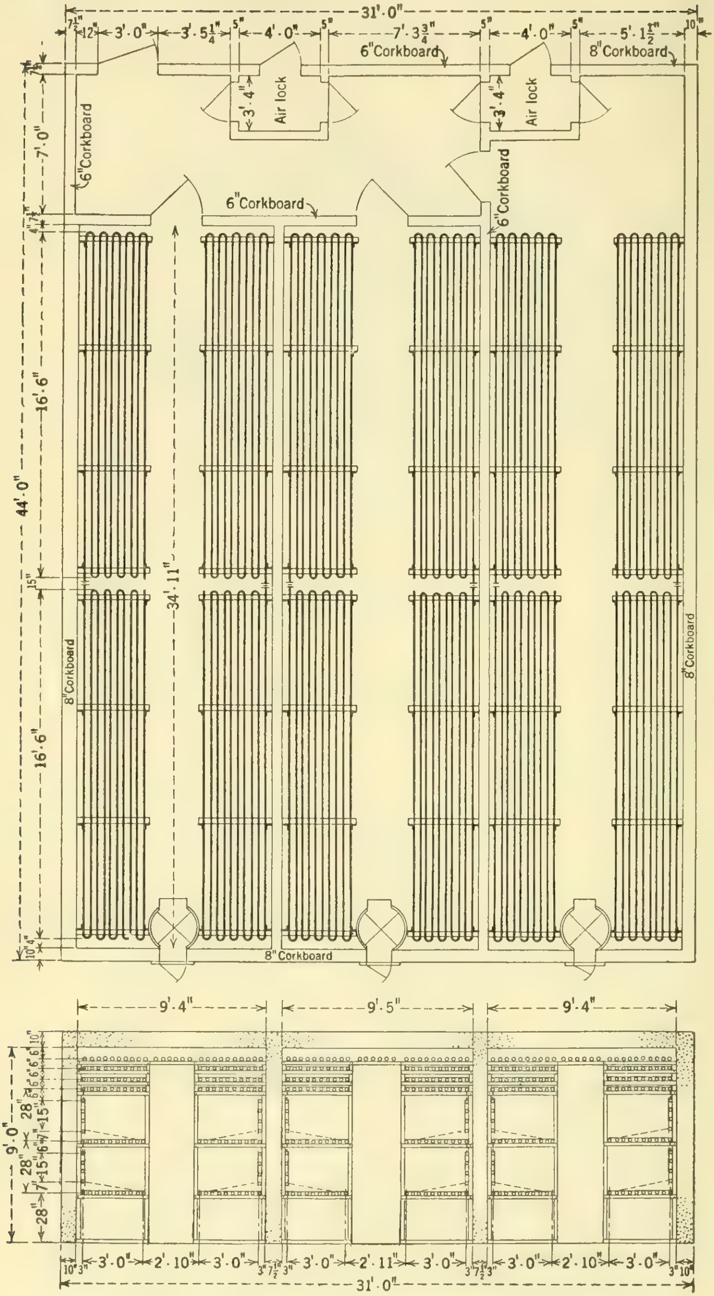


FIG. 279.—The Sharp Freezer.

CHAPTER XIV

COLD STORAGE

The most vital application of mechanical refrigeration is cold storage. The space in the United States cooled mechanically to 40 deg. F. or lower for all kinds of cold storage work, exclusive of the packing house, is estimated at about 700,000,000 cu. ft., and this represents about $250,000,000 of invested capital. No estimate of the amount of goods handled in this space is accurately available. There are few industries so vitally concerned in the life of the nation, touching as this does the staple foods such as meat, poultry, fish, vegetables, fruits, eggs and dairy products.

THE COLD STORAGE INDUSTRY (1925)

By A. W. Oakley

Some thirty-five years ago the cold storage industry carried along in a limited way using natural ice as a refrigerant, but in the past twenty-five years there has been rapid development in the science and art of refrigeration and its adaptation in a commercial way to the preservation of food products from the producer to the consumer. This is indicated by figures giving the approximate refrigerated space in public cold storage warehouses, which in 1914 was 200,000,000 cu. ft. and in 1923, 298,000,000 cu. ft., as given by the United States Department of Agriculture, whereas in 1922 the combined public and private warehouses, including meat-packing establishments, aggregated 559,000,000 cu. ft., of which space about 25 per cent is freezer and 75 per cent cooler. In this space there is carried approximately 1,930,000,000 lb. of apples, cheese and eggs at cooler temperatures; 275,000,000 lb. of butter, poultry, and frozen eggs at freezer temperatures; 75,000,000 lb. of frozen fish and about 1,140,000,000 lb. of meat and packing-house products carried in the cold storage warehouses of meat-packing establishments making a grand total of 3,420,000,000 lb. of products mentioned. These figures do not include such products as dried fruits, nuts and miscellaneous fruits and vegetables and other articles which are not reported by the United States Department of Agriculture, and for which no statistics are available but which represent a very considerable amount. This, with the constantly increasing demand of the small individual refrigerator, as used by the retailer, made possible by automatic machinery, serve the marketing and distribution of perishable foods.

Cold storage warehouses serve to stimulate production and prevent gluts in the market, and the modern warehouse is constantly striving for better handling and preserving of perishable goods in the cold storage rooms by proper use of ozone, ventilation, air circulation and control of humidity. Cold storage warehouses are a necessity as a reservoir in feeding the population in big cities, as some 90 per cent

of the food comes from other states and distant countries. Some idea may be obtained by the following carloads of perishables consumed in New York City per week:

Dairy Products	800
Milk	2200
Meat	750
Live Stock	600
Poultry	300
Fruit	850
Vegetables	1750
Sea Food	50
Total	7300

These commodities are handled by some 60,000 distributors and 400,000 employees. In New York City, with its congested business section, one office building alone housing 10,000 business people, most of whom consume one meal near business, means the providing of 30,000,000 meals per day by New York markets.

The cold storage warehouse also provides against food shortage in emergencies such as railroad strikes and embargoes.

The object of cold storage is an economic one. It provides a means of extending the season and continues the market for vegetables like celery, lettuce, onions, potatoes, cabbages, etc.; for fruits, like apples, pears, plums, oranges, lemons, etc.; for butter, cheese and eggs, and for raw and cooked meats. The purpose of the industry is to retard the natural processes of decay, this delay being accomplished best by freezing, protecting the surface (such as with fish) by a glaze of ice, or by holding the article as close to the freezing point as the nature of the commodity will allow: as for example, apples, which may be kept at 28 or 29 deg. F. Eggs are a glut on the market in the spring, and are scarce and expensive in the fall and the winter. Fruits like plums, strawberries, loganberries, etc., must be canned in order to extend the season over the natural one of a few weeks, unless cold storage can solve the problem.

To serve a useful purpose, then, the cold storage warehouse must take the surplus supply—which must be of the best quality always—and hold it until the demand increases. This means that only under exceptional conditions would the duration of the storage exceed 9 to 10 months, and according to the United States Government reports the average length of storage for various articles is:

	Months
Eggs	5.91
Dressed poultry	2.42
Butter	4.43
Beef	2.28
Mutton	4.45
Pork	0.88

In addition, it is estimated that apples are kept about 5 months, fish (frozen) and dried fruits approximately 6 months.

To give an idea of the relative quantities, it is estimated that the average large cold storage warehouses in the larger cities devote one-quarter of the space to egg storage during the egg season, about 10 to 12 per cent to meats and fish, one-quarter to one-third to fruit storage, and about one-fifth to one-third to butter and cheese. The remainder of the space in the *public* cold storage warehouses is used by a large miscellaneous variety. At special periods, of course, these proportions are very different, as, for example, apples may take half of the space in the fall, and eggs half of the space in the spring.

The Apple.—Although apples do not freeze until the temperature is close to 27 deg. F., it is recommended[1] that the temperature be not lower than 30 deg. F., and that 30 to 31 degrees gives the most satisfactory results, both as regards the holding of the fruit and the condition following the removal from cold storage.

The humidity of the air in the cold storage room is very important (for apples and all commodities held in storage) and only under exceptional conditions is the air too moist. As a rule the *wilting* of the fruit due to the drying out of the moisture is one of the most serious problems, but packing in air-tight boxes or barrels or wrapping each apple will decrease this loss. The usual relative humidity of 80 to 90 per cent with an average of 85 per cent gives good results.

According to Dr. Magnus, United States Government research as to the effect of *fresh* air ventilation has shown no advantage from this. In tests there was apparently no difference in the rate of ripening, in the appearance of the fruit, or in the quality of the apples so ventilated as compared with similar lots not so ventilated. There was no appreciable difference in the amount of sugar or amount of acid, and it was impossible to distinguish any difference in the aroma in such fruit on removal from storage.

In order to prevent scald,[2] however, there must be a lively *circulation* of air. This means that air must be able to reach every individual apple and carry away the heat generated by the chemical action which is always going on to a greater or lesser degree. Scald is caused apparently by the accumulation of certain gases given off by the apples themselves, and it can be prevented only by air circulation or by the absorbing of these gases. An oiled (not waxed) wrapper with 15 per cent by weight of oil decreases scald to a marked degree. Apples scald far less in crates

[1] Dr. J. R. Magnus, Handling of Apples and Pears in Storage, U. S. Bureau of Agriculture.

[2] Farmers Bulletin, No. 1160, U. S. Department of Agriculture.

and boxes, while barrels with 15 holes $\frac{3}{4}$ in. by 4 in. cut in the staves decreased the scald by permitting the air to circulate freely provided the boxes or barrels are properly stored. If the apples are in boxes they should be at least $\frac{1}{4}$ in. apart, and have 1-in. strips between layers. If the apples are in barrels they should be down and not on the ends. The best practice is to stow 6 to 8 high using 2-in. by 2-in. strips at the heads to stimulate natural circulation. Apples are not improved by moving after stowing until ready to ship. The amount of carbon dioxide permissible is not definitely known, but according to experiments in Cambridge, England, 12 per cent of carbon dioxide was found to be an aid in the keeping of the apples, but more than 12 per cent decreased the holding power.[3] British practice advocates the use of the *ozonator* although this is not generally employed in the United States.

The Pear.—According to Dr. J. R. Magnus, the best storage temperature of the pear is 30 deg. F. Handling, storing, air circulation and humidity are practically the same as for the apple. Rapid cooling of the fruit does not injure it, in fact the more rapid the better. Blackening of Bartlett pears is shown by experiment to be due to their being picked too early, in consequence of which there is not the natural skin wax which is a protection. Blackening of pears and scald in apples appear to be similar conditions. In Bulletin No. 377 of the University of California, by Overholser and Latimer, the results from investigations concerning the effect of temperature, the degree of maturity, region where grown, etc., on scald and blue mold are given.

The Egg.—According to Bulletin No. 729, United States Department of Agriculture, only large eggs that are clear and have clean whole shells should be used for long holding in cold storage. The net weight of a case (30 dozen) of eggs packed for storage should be not less than 42 lb. They should be packed in new, odorless cases fastened preferably with cement-coated nails with medium heavy fillers. Odorless fillers should be used on the top and bottom, and the lid should be securely fastened. (See Table 97 for operating conditions.)

The egg room should be dry, clean and free from odors, and the temperature should be kept at from 29 to 32 deg. F. In storing, ventilation must be provided, using 2-in. strips on the floor and $\frac{1}{2}$-in. or thicker strips between the cases in the stacks. If the eggs are fresh and have clean whole shells they may be successfully preserved for from nine to ten months. The loss in spring-packed eggs by candling is about 3 to 4 per cent., but it is much greater for summer-packed eggs. The loss in weight due to shrinkage per case is about 3 to 4 oz. per month. The relative humidity of the room should be about 85 per cent.

[3] A. H. Ashbolt (Agent General for Tasmania), Cold Storage, 1923.

In the summary of Bulletin No. 775 (United States Department of Agriculture) it is stated that clean whole eggs that have not been wet show a negligible loss after 10 to 11 months in storage. Imperfections in handling, grading and marketing previous to storage are responsible for most of the bad eggs. Commercial selection by inspection is stated to be very inefficient. Average tests showed 17½ cracked and one leaking egg per case. Dirty and spoiled eggs were included. Candling is by far the preferred method, and by its means the egg can be graded according to quality, and cracked eggs can be detected. There usually are about 3 eggs cracked per case of spring firsts. If eggs are dirty, cracked, leaking, heated or stale they should be packed in bulk and frozen, under which condition they will keep a year or more. Eggs fresh when stored show after storage an increased air space and a yellowish white. The percentage of ammoniacal nitrogen is an indication of the decomposition. The cold storage taste present after the seventh month seems to be an absorption of the odors from the strawboard fillers or other surroundings.

Dr. Mary E. Pennington,[4] speaking before the American Association of Ice and Refrigeration (1923), said, in explaining certain research in egg storage that aeration with fresh air had a deterrent action on the absorption by the eggs of storage flavors when the present egg package is used and when the temperature carried is from 29 to 31 deg. F. In this research the air was cooled and conditioned by the use of water and brine sprays, and mold appeared to be checked by the use of rapidly moving fresh air, whereas the fact that the air was in a lively circulation did not increase the shrinkage, provided the proper humidity was maintained.

The freezing of eggs in bulk is good economy. It withdraws from the

[4] Dr. Mary E. Pennington, American Association of Ice and Refrigeration, 1923. The conclusion of this paper is as follows:

While the foregoing report is based on but one experiment, and one in which June instead of April or May eggs were under observation, findings indicate that:

1. Aeration with fresh air had a deterrent action on the absorption by the egg of storage flavors when the present standard egg package is used, and when the temperature is from 29 deg. to 31 deg. F.

2. The humidity of the storage rooms was controlled during the late summer and winter months by a regulated current of outdoor air.

3. Apparently moving air does not accelerate the usual shrinkage of stored eggs, provided the air be sufficiently humidified.

4. It is of interest to note that a well started growth of "whiskers," and a beginning growth of mold, was checked by a rapidly moving current of fresh air.

5. The total loss in weight of the eggs averaged approximately 0.5 per cent per month. This is very similar to the loss observed by investigators in the U. S. Department of Agriculture studying the commercial preservation of eggs by cold storage.

market surplus quantities of edible eggs of a grade below first, and makes them available for use the year round. They are put on the market in the form of whites, yolks, or a mixture of whites and yolks. The 30-lb. tin is the usual package. Frozen eggs should be frozen promptly, and should then be kept at a temperature of 10 deg. F. or lower.

Butter.—Butter should be carried at 2 deg. F. or lower to secure the best results. It is usually packed in the 63-lb. tubs, the 63 to 78-lb. cubes and the standard boxes for the one pound prints. Butter should be made of cream of limited acidity if it is to be stored for several months, should be pasteurized, cooled promptly and churned without further ripening. Low percentages of salt in the butter are better for keeping than high percentages. Stowing is important as regards air circulation and inspection. The cube and box packages should be separated by at least one inch dunnage. D. C. Dyer says,[5] that oxidation of the fat itself is not the cause of undesirable flavors in cold storage butter held at zero deg. F., but is due to a chemical change in one or more of the non-fatty substances found in the buttermilk.

Poultry.—Dressed poultry, properly prepared for storage, fresh, absolutely free of any visible or olfactory signs of decomposition and correctly handled may be maintained in good quality for 12 months. It is not advised that longer periods be used, although it is possible, except that the palatability is decreased.

For the best results the poultry should be dry picked and dry packed. Poultry which has been in contact with ice should never be stored. Ice-packed poultry and scalded poultry usually age more quickly. Poultry should be packed in clean, well-made boxes—usually 12 birds to the box—in either single or double layers. Each fowl should be wrapped separately and the box lined with a suitable paper. The box should be tight and the packed boxes should be placed in the sharp freezer and stored so as to permit air circulation. During freezing the room temperature should be from 0 to 5 deg. F., and afterwards it should be held at 15 degrees or lower.

Fish Storage and Freezing.—Fish is one commodity which must be frozen in a special manner in order to be kept in storage for any length of time without serious change of food value or flavor, yet without freezing fish would be practically unknown in inland communities, and such fish as bluefish would be in the market for a few weeks only, while salmon would be unknown, except on the sea coast, in any way but as cured fish. The market for mackerel, halibut, pike, smelts, etc., would be curtailed considerably, and fish of many varieties would be a costly delicacy.

[5] Journal of Agricultural Research, September, 1916.

For storage fish needs ice "glazing." By this is meant that a protective coating of ice be formed on every individual fish or around a number of fish if the fish are small. Unless they are glazed the skin is likely to turn white[6] and the fish to shrivel, because of the loss of moisture. The glaze also tends to prevent the shrivelling of the noses and the fins, to keep the eyes from becoming opaque and the gills (normally red) from becoming dark and brown, and in general, provides a surface on which molds and fungi cannot grow.

As a rule the following procedure is considered good practice. The fish are first washed free of dirt and slime and the larger fish—bluefish particularly as they are heavy feeders—are gutted before freezing. On the contrary, some localities prefer salmon to be frozen in the round. The smaller fish, as the small weak fish, mackerel and salmon, cannot be gutted and are therefore frozen in their natural state in pans containing approximately 40 lb. Large halibut and salmon are frozen separately. The pans are placed directly on the shelving made up of piping (Fig. 279) and the piping ratio is sufficient to permit a temperature of the sharp freezer of from − 5 to − 15 deg. F. The shelving is designed to permit passageway for convenient handling. The fish are reduced in temperature in from 12 to 30 hours, depending on the room temperature and the size of the fish, and as soon as this is done they are ice glazed. The glazing operation is done by having a trough of water at about 32 deg. F. (which is frequently changed) in a room at a temperature of from 20 to 25 deg. F., and the fish are made to pass through the trough. The process is repeated from three to five times or until the required thickness of ice is frozen upon the fish. Another method is to use a grilled platform which is lowered and then raised with the fish. Finally the fish are placed in the ordinary type of cold storage room until used, the room being kept constant at some temperature from 0 to 10 deg. F. After 3 to 5 months the ice glaze will be lost to the air in the room by evaporation, and will need to be replaced. Boxing of the glazed fish is becoming more general, as it simplifies the handling and the shipping of the fish, and permits a more careful record to be kept of the duration of storage. The boxes are lined with heavy Manilla paper, and when so done the fish are protected from injury and the glaze takes longer to evaporate. As a rule 4 to 5 cakes of panned fish, 120 to 150 lb., commonly are packed in one box, the size of which is made just right to take the cakes from the pan. Large fish are usually wrapped first in a vegetable parchment paper, and then are carefully packed in boxes lined with Manilla paper.

Tests by the Bureau of Chemistry, United States Department of

[6] Bulletin No. 635, U. S. Department of Agriculture, by Clark and Almy.

Agriculture (Bulletin No. 635) indicate that properly glazed fish were kept for 27 months during which time and at the end of this period elaborate analysis did not indicate any changes which rendered the fish unsuitable for food, nor were there any important differences in the chemical composition between the stored and the fresh fish. Other experiments show that after 9 months there was nothing in the taste by which the average consumer of fish could detect the stored fish from the fresh fish. However, the average period of storage is about 8 months, as the stock is usually depleted before the fresh fish is again plentiful in the market.

The Brine Method.—According to a paper by Dr. Walter Stiles of University College, Reading,[7] the use of cold brine for the freezing of fish was found to be better than freezing by air cooling. All air cooling, according to this authority, caused a loss of surface sheen and brightness due to the loss of surface mucilage, as well as a loss of weight by evaporation of moisture up to 6 per cent. These changes in the fish were not present in the case of brine freezing. The report goes on to show that an ice glaze of 10 per cent increase of weight should last 30 days, but that brine freezing tends to prevent the formation of this ice glaze. J. M. Tabor[8] says that fish plunged into cold brine will freeze in half an hour, that the brine is found to be an antiseptic, and that the freshness of the fish is not affected because the skin is frozen first. Mr. Tabor advises that the fish be cleaned and gutted before freezing. In packing fish "salunol ice" (ice made from water with a trace of sodium hypochloride) was advocated because of its germicide properties.

The freezing of fish in the United States with brine (a patented process) has not become general. The salt does penetrate somewhat into the body of the fish and at times fish so treated is claimed to be unpalatable. The eyes become dulled, and if blood is exposed to the brine it is changed to a dark color. The fish usually have distorted shapes, and pack poorly. But the process is a commercial one and certainly has merit from the theoretical point of view.

The Peterson Process.—Mr. J. W. Peterson has developed a process[9] which greatly resembles the manufacture of ice. The smaller fish are placed in scoops or pans of about 28 in. by 18 in. by $3\frac{3}{4}$ to 2 in. deep and then are carefully transferred to No. 15 gage galvanized cans (Fig. 280). These cans are submerged in calcium chloride brine at from -20 to -25 deg. F., and complete freezing is possible in 2 to $2\frac{1}{2}$ hours, or less. The frame of cans is then lifted out of the brine and is lowered

[7] Report of the Food Investigation Board, 1921, England.

[8] J. M. Tabor, British Cold Storage and Ice Association, 1923.

[9] J. W. Peterson, Refrigeration Engineering, 1924.

into a thawing tank (as in ice making) and when the "cake" is released it is glazed as before. Large fish are packed into the cans individually. It is claimed that the Peterson process offers a low first cost, a small space requirement, that the freezing time is very short, and also that the power and the labor costs are comparable with other methods of fish freezing.

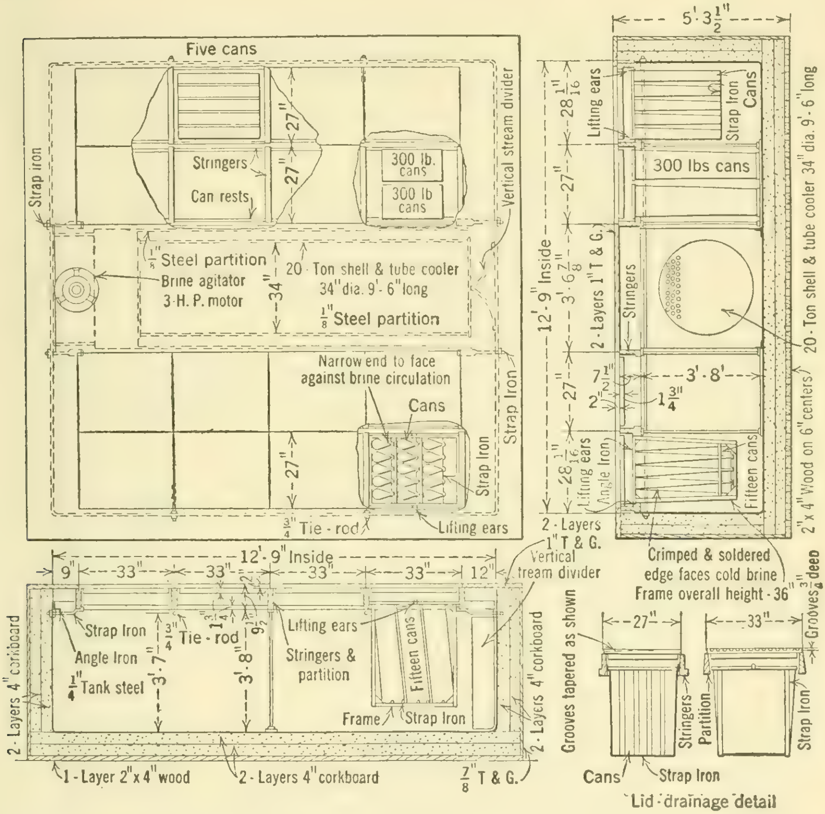


Fig. 280.—Peterson Tank System for Fish Freezing.

Vegetables: *Sweet Potatoes.*—Sweet potatoes, free from frost, bruises, and dried sufficiently to remove the surface moisture are stored in bins, crates and hampers, stacked so as to permit good air circulation. They should be "cured" at 85 deg. F. to remove surplus moisture (this requiring from 7 to 21 days) at which time the surface is bright, clear and dry,[10] and then the temperature should be gradually reduced

[10] H. C. Thompson, Storing and Marketing of Sweet Potatoes, No. 970, U. S. Department of Agriculture.

from 50 to 55 deg. F. The room should be kept dry, under which conditions the sweet potato will keep well for from 3 to 6 months.

Onions.—Only sound, well ripened onions should be stored—free from loose skins—and properly dried. They should be stored in slatted onion crates, properly stacked for air circulation, or in shallow slatted bins and should be cooled to from 32 to 36 deg. F. as quickly as possible after stowing. A low relative humidity should be carried.

Cabbages.—Solid heads, free from injury and loose leaves, should be placed one layer deep on slatted shelves properly arranged for good air circulation. In order to prevent excessive wilting and shrinkage a

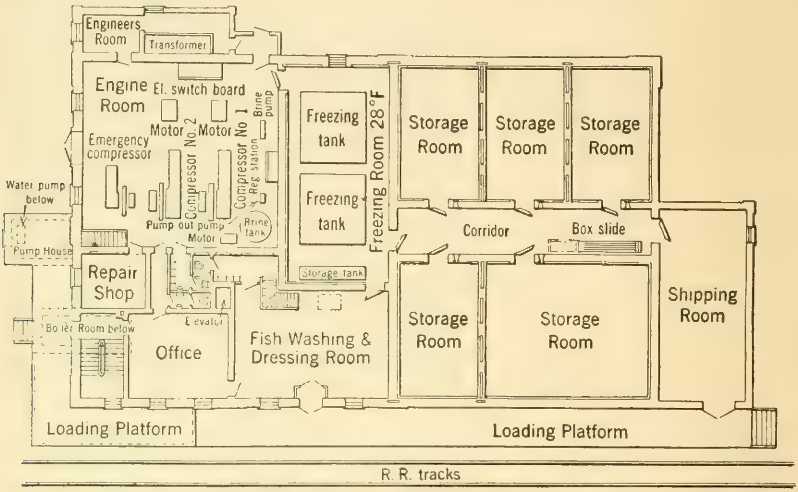


FIG. 281.—The Otteson Fish-Freezing System.

relative humidity of 80 to 85 per cent should be carried, but moisture must not be permitted to form on the leaves.

Potatoes.—Potatoes should not be stored deeper than 6 ft., and if in crates or bags good circulation of the air in the room should be arranged for. The air temperature should not be less than 35 nor greater than 40 deg. F. The relative humidity should be as high as possible without precipitation of moisture on the surface, or 80 to 90 per cent. Potatoes should keep 6 months in prime condition.

Insects in Cereals.—In a paper by Dr. A. E. Black of the United States Department of Agriculture before the American Association of Ice and Refrigeration, 1922, it was stated that the fruit fly is very easy to destroy, but that the clothes moth larvæ and other pests are not so easily killed. The bean weevil can be killed if reduced to 32 deg. F. for

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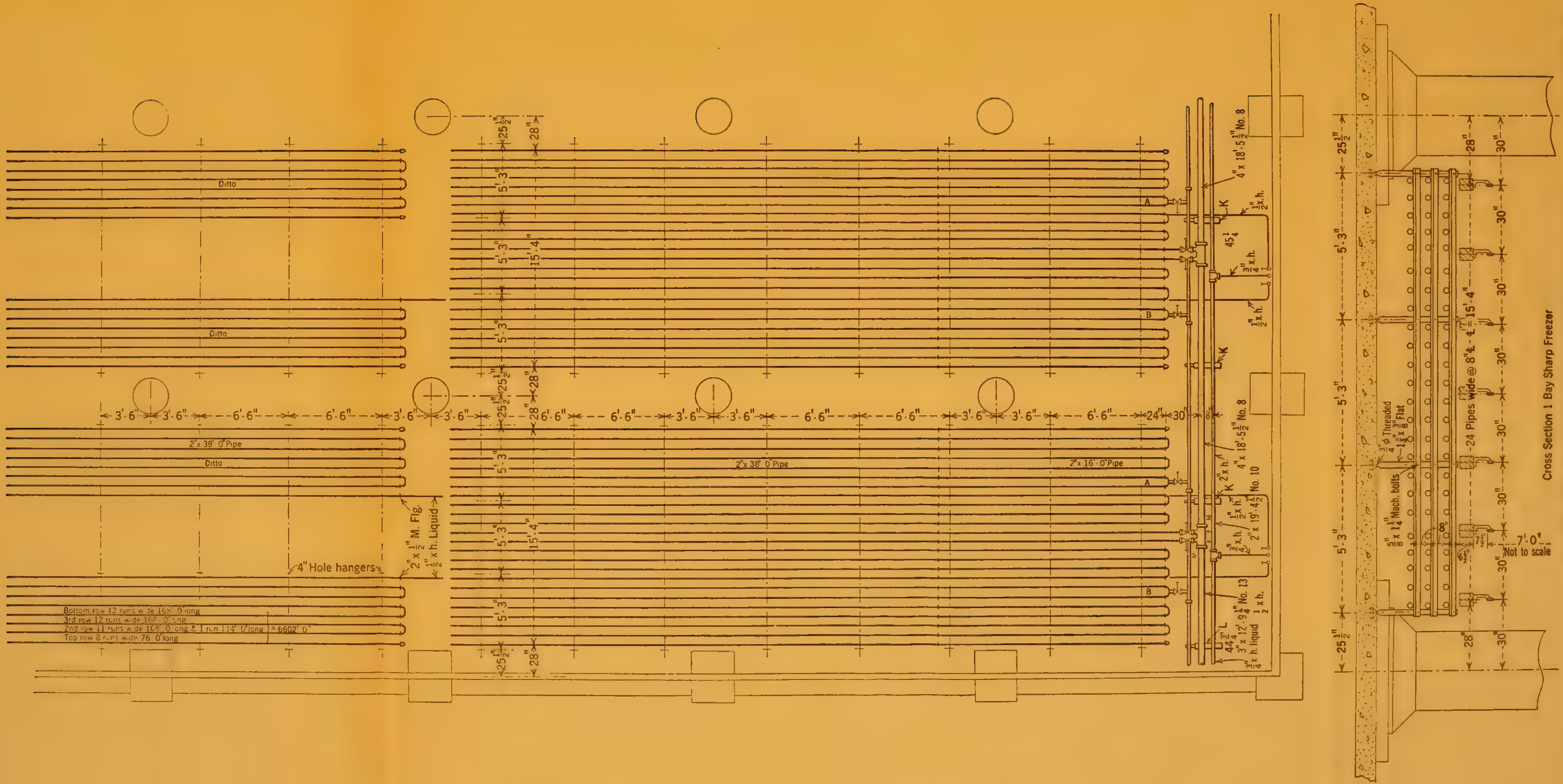


FIG. 282.—Beef Sharp Freezer for Direct Expansion of Ammonia.

56 days, whereas the cow pea weevil will succumb in a month at 32 deg. or even at 39 deg. F. The following insects were subjected to 30 deg. F.

The sawtoothed grain beetle, common in flour and dried fruits, was killed in four weeks. The northern flat grain beetle dies in a week, and the southern variety in two months, whereas the broad-horned flour beetle required only one week. The Australian wheat weevil required from three to four months, as did also the tobacco beetle which likewise attacks cereals. The rice weevil egg was killed in one week, the adults in 17 days and the larvæ in four weeks. The granary weevil (somewhat similar to the rice weevil) required three months at 30 deg. F. The eggs were killed in four weeks and the larvæ in four to five weeks.

Freezing of Fruits and Berries.—Research in the preservation of fruits and berries[11] by freezing at 8 to 12 deg. F. showed that, in general, soft fruits and berries can be " kept for from 6 to 10 months at 8 to 12 deg. F., after which they are equal to fresh fruit for most purposes." Cherries in open containers become dark in color, but covering with water and freezing prevented this darkening. The use of sulphurous acid resulted in an inferior flavor. Apricots gave excellent results, especially when stored in water, or crushed with an equal weight of sugar. Red raspberries, currants, loganberries and strawberries, untreated, preserved well at 8 to 12 deg. F., but shrivelled slightly. Best results were obtained by crushing the fruit and adding sugar. The flavor was much better than similar fruit preserved by cooking. Grape juice stored at 10 deg. F. was superior to the pasteurized variety.[12]

Plums.—E. L. Overholser[12] says that certain varieties of plums keep well at 36 deg. F. and others at 32 deg. F. The storage time averaged from 10 to 12 weeks and gave a marketable plum of from 2 to 7 days. Sorting and packing are prompt. Precooling to 32 deg. F. is imperative.

SOME EFFECTS OF FREEZING ON MATURE FRUITS OF THE APPLE

By D. B. Carrick, Cornell University Agriculture Experiment Station.

Rapidly frozen apples exhibit a larger amount of discoloration than do similar fruits given four times as long an interval to reach the same minimum temperature. Apples frozen rapidly to varying degrees, when thawed at 0 deg. C. (32 deg. F.) and 22 deg. F. (72 deg. F.) show an equal amount of browning injury. However, slowly frozen apples when thawed very slowly during several days or weeks at 0 deg. C. (32 deg. F.), uniformly reveal more discoloration than if more quickly thawed at 10 deg. C. (50 deg. F.) or higher.

[11] Cruess, Overholser, and Bjarnason, Bulletin 324, University of California Agriculture Experiment Station.

[12] E. L. Overholser, Bulletin No. 344, Agriculture Experiment Station, University of California.

Freezing points of expressed apple-fruit juice, measured by a Beckmann apparatus, are found to be significantly higher than the depression of the cell sap within the normal, unfrozen apple tissue as determined by thermo-junctions. The depression of the cell sap within the tissue frozen to death was much higher than the Beckmann measurements.

Various external and internal symptoms of frozen apples, such as changes in color, the effect of bruising while ice is still present, and the texture and flavor, are described. Some distinctions are made between frozen and over-mature apples.

Since slightly frozen apples are usually more susceptible to *Penicillium expansum* and other fungous organisms, they do not keep so well in storage as does normal fruit. Severely injured apples cannot be held without loss for any length of time, and should either be disposed of immediately or placed in storage at − 20 deg. C. (− 4 deg. F.). In this low temperature the fruit can be kept almost indefinitely, and it is satisfactory as a culinary product but worthless for fresh consumption.

Preliminary results with the Wagener, Baldwin, Rome, and Ben Davis varieties indicate that − 1 deg. C. (30.2 deg. F.), or somewhat below this point is the optimum cold storage temperature for the apple, as compared with 0 deg. C. (32 deg. F.) or above.

A study of extreme temperature is contained in the United States Department of Agriculture Bulletin No. 1133, entitled, "Freezing Temperatures of Some Fruits, Vegetables, and Cut Flowers," by R. C. Wright and Geo. F. Taylor.

Freezing or freezing injury does not always occur when fruit or vegetable products are exposed to temperatures at or below their actual freezing points. Under certain conditions many of these products can be undercooled; that is, cooled to a point below the true freezing temperature of each and again warmed up without freezing and without apparent injury. Certain products under certain conditions may be actually frozen and then thawed out without apparent injury, while, on the other hand, some products are injured if stored at temperatures well above their actual freezing points. Evidence seems to show that different individuals of the same variety and strain when grown under different conditions will have somewhat different freezing points, and that there are also some variations in the freezing points of products of the same variety and from the same lot.

In view of these facts the freezing points given in this bulletin should be considered only as danger points at or near which, either above or below, there is a possibility of freezing injury if exposed for a sufficient length of time. These are temperatures at which it is unsafe to hold produce for any length of time, as serious danger of frost injury exists.

Cold Storage for Furs and Fabrics.—Because of the damage from the ravages of moths and beetles, the fur storage industry has developed to quite an extent during the summer months. Storage at freezing temperatures also tends to improve the lustre of the fur so that the fur has a greater value after the warm weather is over and the furs are needed agan for the fall or the early winter.

Temperatures Required.—The moth and the beetle egg requires a temperature of 55 deg. F. for hatching. If hatching occurs the insect

eats the fur to secure the grease and the animal juices existing there. The fur constitutes the chief food of the insect and supplies the necessary material for the cocoon of the moth. If the larvæ are hatched the movement becomes active at 45 deg. F. and above, and the fur will be injured; this activity increasing up to 55 deg. F. where normal activity is reached. Temperatures below 45 deg. F. stunt the activity to a point where little or no damage is done; at 42 deg. it has been demonstrated that although the insect may be still moving, it is very sluggish; and at 40 deg. F. all movement is suspended.

TABLE 93

SUMMARY OF AVERAGES

Fruit and Varieties	Temperatures, Degrees F.		
	Average	Extremes	
		Minimum	Maximum
Apples:			
Summer varieties	28.44	28.12	28.62
Fall and winter	28.51	28.21	28.87
Bananas (Jamaica):			
Green { Peel	29.84	29.76	29.92
Green { Pulp	30.22	30.10	30.58
Ripe { Peel	29.36	29.15	29.53
Ripe { Pulp	26.00	25.45	26.50
Blackberries:			
Black varieties	29.15	28.73	29.42
White varieties	28.40	28.12	28.63
Logan (*Loganberry*)	29.51	29.32	29.75
Cherries	27.81	27.56	28.25
Cranberries	26.70	26.28	26.93
Currants	30.21	30.18	30.25
Gooseberries	28.91	28.70	29.18
Grapes (eastern)	28.16	27.85	28.37
Grapefruit	28.36	28.00	28.50
Lemons	28.14	27.89	28.47
Oranges	28.03	27.86	28.34
Peaches (hard, ripe)	29.41	29.09	29.74
Pears (Bartlett):			
Hard, ripe	28.46	28.06	28.70
Soft, ripe	27.83	27.20	28.00
Pears (unknown Japanese variety)	29.39	29.34	29.53
Japanese persimmons (Tanenashi)	28.33	28.07	28.63
Plums	28.53	28.20	28.85
Raspberries:			
Red varieties	30.41	30.12	30.50
Black varieties	28.76	28.24	28.79
Strawberries	29.93	29.56	30.13
Kind and Variety			
Beans (snap)	29.74	29.65	30.06
Cabbage (Early Jersey Wakefield)	31.18	31.06	31.34
Carrots	29.57	29.42	29.68
Cauliflower	30.08	29.95	30.15
Corn, sweet	28.95	28.65	29.22
Eggplant	30.41	30.17	30.69
Kohl-rabi	30.02	29.74	30.22
Lettuce	31.20	31.03	31.38
Onions (dry)	30.09	29.69	30.24
Peas (green)	30.03	29.67	30.25
Potatoes	28.92	28.80	29 02
Potatoes, sweet	28.44	28.10	28.72
Tomatoes (ripe)	30.38	30.20	30.67
Turnips	30.23	30.16	30.48

TABLE 94

Product	Handling Factors: Pre-storage handling and condition when placed in storage	Handling Factors: Varieties and grades	Warehouse Requirements: A—Construction	Warehouse Requirements: B—Temperature	Warehouse Requirements: C—Humidity	Warehouse Requirements: D—Stowing
Apples....	Apples should be picked when well matured but not over-ripe. In all the operations of picking, packing and hauling they should be so handled as to avoid bruising, skin punctures and other mechanical injuries; and they should be so graded as to be practically free from serious injuries caused by insects, diseases or mechanical means. It is essential that they be handled promptly from the orchard to the storage room and cooled quickly.	Only varieties which have a recognized storage period of three months or more should be considered.	Cold storage houses should be so constructed and equipped as to maintain practically uniform temperature and humidity conditions throughout the storage season. Common storage houses should be sufficiently insulated to prevent freezing, and should be provided with the necessary inlet and outlet vents to permit adequate ventilation and temperature regulation.	Cold storage temperature range should be 31 deg. F. to 32 deg. F. for the storage of apples. Common storage temperature should be maintained at from 31 deg. F. to 36 deg. F. after the initial cooling of the fruit.	Humidity range, 80 to 90 per cent.	Apples should be stowed with sufficient spacing to permit of free air circulation, and to render each lot readily accessible for inspection and withdrawal.
Potatoes..	Potatoes should be well matured, and graded to conform to the specifications of the United States standard grades. Seed stock should be certified by a competent inspector.	All varieties harvested in autumn keep well in storage.	The building should be so constructed and insulated as to prevent fluctuations in temperature. Means for ample ventilation should be provided and all unnecessary light should be excluded.	Temperature range from 35 deg. F. to 40 deg. F.	Humidity range, 80 to 85 per cent.	When stowed in bags, boxes or crates, potatoes should be so piled as to permit free air circulation. Bulk potatoes should not be stored to a greater depth than 6 ft. nor more than 60,000 lb. in a single compartment. They should be carefully handled to avoid unnecessary injuries.
Potatoes (sweet)..	Sweet potatoes should be well developed, carefully handled to avoid bruising, and should be practically free from damage caused by disease, insect or mechanical injury. They should not be allowed to become chilled or frosted, and when placed in storage the surface should be dry and practically clean.	All varieties grown on a commercial scale.	The building should be so constructed that all light is excluded, and moderate changes in outside temperature will not quickly affect inside temperatures. Wood construction is preferable, and ample means for ventilation control should be provided.	While the potatoes are being stored, and for a period of ten days to two weeks thereafter, or until the potatoes are cured, a temperature of from 80 deg. F. to 90 deg. F. should be maintained. Thereafter, a uniform temperature of as nearly 55 deg. F. as is practicable should be maintained. Ventilation and artificial heat are necessary to control temperature and moisture.	The percentage of humidity should not be so high that moisture is deposited on the walls of the storage house.	When stored in bins, the potatoes should be carefully poured from basket or crate into the bin. To allow free circulation of air, the bins should have slatted sides and floor, and at least 4 in. of air space on all sides. The bin floor should be raised 2 in. or more above the house floor. When stowed in crates, baskets or hampers, the containers should be stacked so as to allow circulation of air, and to avoid the crushing or breaking of the packages and the bruising of their contents.
Onions...	The onions should be well ripened, dry and thoroughly air-cured when stored. Onions intended for storage should be practically free from damage caused by disease, insects or mechanical injury, and from other stock commercially known as culls.	All common varieties of onions, except those of the Bermuda type.	The building should be so constructed and insulated as to prevent fluctuations in temperature, and means for ample ventilation should be provided.	In cold storage the temperature range should be from 32 deg. F. to 36 deg. F. In common storage the same range of temperature should be maintained as nearly as possible, but with the proper ventilation, onions will keep well in common storage at higher temperatures.	Low humidity is desirable.	Onions should be stowed in suitable receptacles, as indicated under "Containers," and should be stacked in such a way as to permit of free air circulation throughout the lot.

Cabbage..	Cabbage must be of solid heads, practically free from injuries caused by insects and diseases. Heads should be cut with but few, if any, loose leaves adhering, and carefully handled from field to storage house. Special care should be used to avoid bruising and other mechanical injuries.	Danish Ball Head, or sorts with similar form and texture.	Well ventilated, frost proof root cellar or warehouse type of construction, with ample intake and outlet vents for quick cooling and ventilation, and equipped with slatted shelves supported on staging, so that the heads may be stored one layer deep, with at least 15 to 18 in. clear space around the walls of the building. The ceiling should be so constructed as to prevent drip on the product.	Temperature range, 32 deg. F. to 36 deg. F.	The humidity should be maintained as high as possible without actual deposition of moisture on the product.	Cabbage should be stored on slat shelves in single layers. The height of the staging and the number of shelves will be determined by convenience and dimensions of the building.
Eggs......	Eggs should be moved quickly from the producer to the warehouse. They should be carefully sorted and candled, so that none showing mechanical defects or noticeable deterioration is included in the storage stocks. No washed eggs should be stored.	The grades should conform to those generally adopted by the wholesale trade, until United States standards are promulgated.	Cold storage houses should be so constructed and equipped as to maintain practically uniform temperature and humidity conditions required for successful storage throughout the storage season.	Temperature range, 29 deg. F. to 32 deg. F.	Humidity range, 82 to 85 per cent.	Egg cases should be stowed so that separate lots may be easily inspected, and with ½ in. to 1 in. dunnage between the cases to insure space for free air circulation.
Frozen eggs	Eggs should be removed from shell in chilled, sanitary surroundings, and frozen immediately on fish-shelf sharp freezers.	One grade for food. One grade for manufacturing purposes.	Same as for eggs.	Temperature range, 0 deg. F. or below to 10 deg. F. above.	The usual humidity at the temperature of storage.	Protect eggs from heat leakage at doors and elevator shafts.
Poultry...	Poultry should be dry picked, dry cooled, and dry packed at temperatures ranging from 30 deg. F. to 35 deg. F. for from 18 to 24 hours, then frozen at 6 deg. F. or below.	The classes and grades should conform to those generally adopted by the wholesale trade, until United States standards are promulgated.	Same as for eggs.	Preferred temperature, 0 deg. F. to 10 deg. F. Admissible temperature, 12 deg. F. to 14 deg. F.	Same as frozen eggs.	Poultry should be so stowed that separate lots may be inspected easily, and protected from injury by heat leakage at doors and elevator shafts.
Butter....	Butter should be placed in cold storage within ten days after it is manufactured. When storage facilities are not available during this period, the product should be held in a temperature below 40 deg. F.	The grades should conform to those generally adopted by the wholesale trade, until United States standards are promulgated.	Same as for eggs.	Temperature, 2 deg. F. or below.	Same as frozen eggs.	Packages of butter should be so stowed as to permit a free air circulation beneath the pile, and so stacked that separate lots may be inspected easily. Cube and box packages should be separated by 1 in. dunnage.
Fish......	Fish should be placed in storage in a fresh condition, as indicated by their physical appearance.	Practically all kinds used for food.	Same as for eggs.	Hard frozen and glazed at temperature of −5 deg. F. or below, and stored at 0 deg. F. or below to 10 deg. F., depending on the kind. For holding less than six months it is admissible to store at 12 deg. F.	Same as frozen eggs.	Fish should be stowed as compactly as possible.

TABLE 94—*Continued*

Product	Warehouse Requirements		Other Considerations		
	E—Containers	F—Inspection	Storage period	Shrinkage	Remarks
Apples....	Containers shall be clean, strongly built barrels, boxes or crates, and when packed for market shall be plainly marked with the grade variety and the grower's or packer's name.	All lots of apples should be inspected when received for storage by a qualified inspector. Subsequent inspections of representative packages of all lots should be made at intervals of 15 to 30 days, depending on the variety and condition of the fruit as indicated by previous inspections.	The usual cold storage period for winter varieties of apples is from three to six months, depending upon the variety and condition of the fruit when stored.	The shrinkage in cold storage is from 2 per cent to 5 per cent. In common storage the shrinkage is variable.	Attention is directed to the fact that a delay of one or more weeks between the picking and storing of apples greatly reduces the storage period of the product and results in early deterioration. The successful storage of apples is as much dependent upon the treatment they receive before being placed in cold storage as the conditions under which they are held in storage. See Department Bulletin No. 5877.
Potatoes..	Potatoes may be stored in clean burlap bags, barrels, boxes or crates; or when in bulk they should be stored in ventilated bins.	Potatoes should be inspected by a qualified inspector when received for storage, and again within 30 days. The frequency of the inspections thereafter will depend upon the condition of the potatoes as determined by previous inspections.	The usual storage period is from three to six months, depending upon the section of the country in which the storage is located, the type of the storage house and the condition of the stock. Allowing for a high percentage of deterioration, potatoes may be held in storage for a much longer period.	When potatoes are stored, in containers or in bulk, as specified in Column 6, the shrinkage may amount to about 7 per cent, although it varies greatly.	Potatoes are usually stored in cellars and common storages, but are sometimes held in cold storages. See Farmers' Bulletin No. 847.
Potatoes (sweet)..	Sweet potatoes are usually stored in bins, but may be stored satisfactorily in substantial crates, baskets or hampers which permit of a free air circulation.	The potatoes should be thoroughly inspected by a qualified inspector at the time they are put in the storage house, within 15 days after the beginning of the storage period and from 15 to 30 days thereafter.	The safe storage period is about four months. Under the most favorable conditions and good management sweet potatoes may be kept six months.	The shrinkage from loss of moisture is from 8 per cent to 10 per cent in bins, and somewhat higher in packages. An additional shrinkage of 5 per cent should be allowed for decay.	It is recommended that sweet potatoes be not considered properly stored until they have passed through the curing period. See Farmers' Bulletin No. 970.
Onions....	The best containers are slatted crates, although baskets, hampers and bags are used successfully.	Thorough inspection should be made when the onions are placed in storage, and at intervals not exceeding 30 days. The frequency of the inspections thereafter will depend upon the condition at the previous inspection.	The usual storage period of onions with proper ventilation is six months.	The shrinkage should not exceed 10 per cent or 12 per cent.	See Farmers' Bulletin No. 354.

Cabbage . .	Containers are not generally used.	Cabbage should be inspected at intervals of from 15 to 30 days.	The storage period for cabbage extends from November to April—five or six months.	The shrinkage in cabbage is quite variable.	Stoves should be provided in common storages to prevent freezing in cold periods. See Farmers' Bulletin No. 433.
Eggs . .	Eggs should be packed in clean, odorless, wood cases. Fillers should be of new No. 1 or medium straw or wood pulp board with flats over top and under bottom. Padding must be kiln-dried excelsior, cork shavings or corrugated straw or wood pulp board on top and bottom of each case. No pine excelsior should be used. The cases should be plainly marked with the grade.	Inspection of eggs should be at intervals of from 15 to 30 days, and the storage house should have daily attention from a competent warehouseman skilled in the handling of such structures and commodities.	The storage period for eggs should not exceed 12 months.	The shrinkage depends upon the humidity, and should not be more than 5.5 per cent. Shrinkage should be calculated from net weight of products.	Rooms must be clean and odorless. See Bureau of Chemistry Circular No. 64.
Frozen eggs	Thirty-pound tin buckets are most common. The use of smaller tin cans is now increasing, due to wider use of this product.	Inspections of frozen eggs should be made about every 30 days.	No change in composition up to 24 months. After 12 months egg thickens slightly. Whites near top of can may become pink, due to iron under tin. Egg not injured as food thereby.	The shrinkage is not of commercial importance.	See Department Bulletin No. 51. Department Bulletin No. 224. Bureau of Chemistry Circular 98.
Poultry . . .	All poultry should be packed in clean, strongly built, odorless boxes, lined with parchment or other suitable paper, and should be plainly marked to indicate the grades and classes. Barrels are still admissible, expecially for turkeys, but are less desirable than boxes.	All lots of poultry should be inspected by a qualified inspector when received for storage, and at intervals of 30 days or longer, depending upon the conditions found at the previous inspections.	The storage period for poultry should not exceed 12 months.	The shrinkage varies from 1 per cent to 3 per cent.	Water-cooled or ice-packed poultry should not be stored for long periods. Scalded birds deteriorate more rapidly than dry-picked. Drawn poultry should never be stored. See Bureau of Chemistry Circulars Nos. 64 and 70.
Butter	Packages should conform to the regular commercial styles, including 63 lb. tubs, 63 to 78 lb. cubes and standard boxes of 1 lb. prints.	All lots of butter should be inspected by a qualified inspector when received for storage, and at intervals of 30 days or more, depending upon the quality and condition of the lots at previous inspection.	The storage period for butter should not exceed 12 months.	In general, the shrinkage will run from $\frac{1}{2}$ per cent to 1 per cent.	See Bureau of Animal Industry Bulletins Nos. 84 and 148.
Fish . . .	Fish are stored in boxes and in bulk.	Inspection of fish should be made at intervals of 30 days or more by a qualified inspector.	The storage period for fish should not exceed 12 months.	The shrinkage is not of commercial importance.	Boxed fish should be reglazed in three to six months. Stacked fish should be spray-glazed every three months or more frequently. See Department Bulletin No. 635.

The *moth miller* and the *beetle* are killed at a temperature of 40 deg. F., but relatively slowly, whereas at 32 deg. F. they are killed quickly. If they are rolled up in rugs or garments they may be protected for weeks, but the evidence is that no damage is done under the conditions of 32 deg. to 40 deg. F. In fur or rug storage it is better to hang up the garment and to lay the rug out in flat piles, and the temperature must be 40 deg. or lower. The larvæ may survive a temperature of 18 deg. F. for weeks, but usually it may be killed if the temperature is permitted to rise to 45 to 50 degrees for a day or two and then lowered again. As a rule, 25 degrees is the preferred operating temperature.

EFFECT OF COLD STORAGE ON CLOTHES MOTHS

By E. A. Back and R. T. Cotton, Bureau of Entomology, U. S. Department of Agriculture

Cold storage is the best method of protection against clothes moth damage. Once in cold storage no injury can take place. Cold storage can be depended upon for absolute protection by dealers in furs, carpets and other valuable articles such as stuffed animal heads, blankets, carriage robes, curtains, upholstered furniture, etc. This is true because clothes moth larvæ or worms cannot feed at temperatures below 45 deg. F. A temperature range lower than one of 40 deg. to 42 deg. or 45 deg. F. is unnecessary.

If cold storage will protect, and has the sanction of the United States Department of Agriculture, why is it that warehousing concerns often have complaints lodged against them by patrons who state that living moths' larvæ are found by them in articles several days after removal from cold storage? The presence of living larvæ in such articles can be explained in two ways. First, larvæ may have crawled to the articles after they were removed from storage from other infested material in the home, but this possibility is rather remote if living worms are detected within several days; second, that while cold storage temperatures were sufficient to protect, they were not sufficient to kill. Cold storage concerns should determinine in advance as a matter of policy whether they offer a service of protection against damage for the period of storage, or whether in addition to this they are to guarantee the articles refrigerated to be free from living moths at the time they leave storage.

There is very little exact data on the effect of cold temperature upon the different stages of clothes moths. Some years ago Dr. A. M. Reed, of the Security Storage Company in Washington, D. C., conducted experiments in cooperation with Dr. L. O. Howard, Chief of the Bureau of Entomology, which indicated that larvæ of the webbing clothes moth *tineola bisselliella* can survive for a long time a temperature of 18 deg. F. No explanation was given of what a "long time" meant in actual days. It was found, as has been discovered in the case of other insects, that it isn't always the cold alone that kills, but the sudden fluctuations from a cold to a warmer temperature and back to a cold temperature. These experiments, referred to above, resulted in the present recommendation of the Department that to kill clothes moths in storage the infested articles should be refrigerated at 18 deg. F. for several days, then suddenly exposed for a short time to 50 deg. F., and then returned to 18 deg. F., and finally held permanently at about 40–45 deg. F.

If a concern merely guarantees to protect articles during the period of storage, it is sufficient to maintain a temperature of about 40–45 deg. F. At this range the eggs of the moth which are apt to be on the article at time of storage will be killed if the period of storage is prolonged over six weeks. Experiments have shown that at 20–25 deg. F. and at 25–30 deg. F., clothes moth eggs are all killed in about three weeks; at 30–35 deg. F., in 26 days, and at 35–40 deg. F. in about one month. But the older well-grown larvæ are very resistant. While these well-grown larvæ were killed in 67 days when subjected to an even temperature of 20–25 deg. F., and in 93 days at 25–30 deg. F., others held at temperatures ranging from 30–35 deg. F. and 35–40 deg. F. are still alive after over four months.

The ability of well-grown larvæ to withstand long periods of refrigeration at moderate temperatures was demonstrated recently in a Washington plant whose rooms were said to have had a temperature range of 24–48 deg. F., but with the temperature mostly at about 40 deg. F. Larvæ were removed after storage at the end of 6, 8, 10, 12, and 14 months. Storage for 6 to 8 months had no appreciable effect upon larval mortality. After 10 to 12 months' storage, many larvæ died soon after removal, and only a few survived 14 months' storage. Yet these few were thoroughly normal, and upon being placed in a sufficiently warm temperature for feeding, resumed activity and transformed the pupæ and adults as though their life had not been prolonged by an enforced 14-month hibernating period.

These facts are important for they explain why patrons have at times found living robust larvæ in articles several days after removal from 4 to 5 months in cold storage. If a fairly even temperature around 40 deg. F. has been maintained, such a discovery is to be expected and is no reflection upon the storage firm. Experiments under way will determine definitely the effect of various temperature ranges upon not only the eggs and the well-grown larvæ, but upon the larvæ of different ages.

The fur storage room is not opened to much of any extent after the goods are placed in storage until after the summer is over and the refrigeration required is nearly all "heat leakage." Such storage rooms require no windows of any sort, and in consequence no correction has to be made for window infiltration or leakage. Electric lights are used, of course, but the amount of heat generated by their use is very small after the goods are once stored. Only one door is usually provided, and that is of the usual cold storage construction and is generally tight and of the same insulating value as the rest of the room. The insulation is usually of 5 to 6-in. corkboard, erected in thicknesses of 2 in. with the joints broken and put up in cement mortar or asphalt. A half-inch cement mortar finish is given the inside of the walls and the ceiling.

In providing refrigeration, piping in the rooms is almost never used, except, perhaps, in the very small rooms. If piping is used it should be very carefully protected so that the frostation on the pipes could never drip on to the goods. There is danger also from the pipes should leaks occur, both from the brine and the ammonia. It is not known whether carbon dioxide is innocuous to furs and fabrics, but the understanding is that no appreciable harm occurs. The preferred method of cooling is by the use of a bunker room. With small rooms the air may be admitted

in one or two places, but the larger rooms need distributing ducts for the supply and the return. Such ducts may, of course, be designed for velocities as high as 2100 ft. per min., but as a rule lower velocities are used—from 1200 to 1600 being usual. Refrigeration is different from heating and ventilation as regards the use of fans. The fan not only uses power to drive the air, but the heat equivalent of the work done must be neutralized by the use of the refrigeration machine. Excess power supplied then is a double expense in the case of refrigerating plants, and in fact there is required a $\frac{1}{5}$ ton of refrigeration for every horse power supplied to the fan.

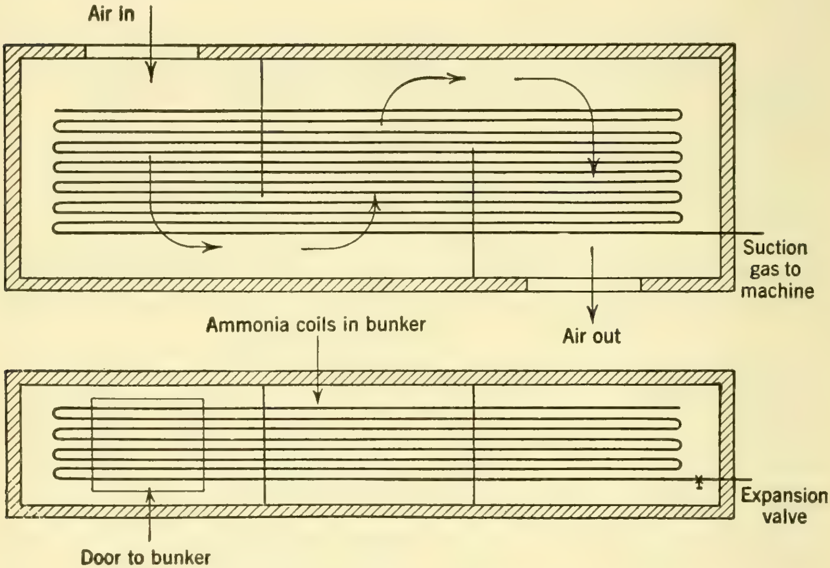


Fig. 283.—Simple Bunker Room Coil for Direct Expansion of Ammonia.

The *bunker* is usually designed (as in the case of the chocolate dipping spray chamber) for 550 ft. per min. air velocity, and the bunker is usually arranged for three passes of the air, and in some of the constructions the baffles are used to catch the condensate and guide it to the waste pipe. Where the size of the plant warrants it, there is no reason why the brine spray chamber cannot be used, with eliminators as in the case of the cooling of air. In fact there are now on the market small spray cabinets which would meet any requirement. As air infiltration is very small, and the commodities do not dry out as do vegetables, fruits and meats, there will be little moisture condensed and therefore very little trouble from diluting of the brine.

The figure (Fig. 283) shows a simple pipe-coil bunker room for ammonia. Ammonia piping is now designed for little or no return bends or other fittings. Instead the pipes are bent and the separate pipes are welded, usually by the electric method. A number of such pipes are welded together (as many as can be handled without difficulty during erection) and similar units are joined together by means of the usual tongue and groove flanges. Or, sometimes the job is made up at the point of erection by the use of thermit welding of different coils, and in this manner a complete welded system of piping is obtained, but thermit welding is much more costly.

TABLE 95

Carload Shipments to Principal Cities

Figures containted in Information Bulletin No. 272, of the Perishable Freight Conservation Bureau, American Association of Ice and Refrigeration, give some idea of the large amount of fruits and vegetables consumed in the principal cities of the United States. While it is not known just what proportion of these cars were refrigerated, it is presumed that a very large percentage of them, owing to the character of the product transported, were under refrigeration. The number of carloads unloaded at the various cities are as follows:

Unloaded at	Carloads
Boston	30,528
Chicago	48,222
Cleveland	12,389
Cincinnati	10,573
Detroit	11,969
Kansas City	7,601
Minneapolis	3,797
New York	94,194
Philadelphia	28,559
Pittsburgh	20,145
St. Louis	12,115
St. Paul	1,891
Washington, D. C.	5,014

In the case of certain commodities the records and reports did not begin until June 1, 1923.

From the foregoing it is evident that the refrigerating load is a small one. It is almost entirely a leakage load, and therefore it depends on the insulation provided. With 5- or 6-in. corkboard the volume of space per ton of refrigeration is about 3000 cu. ft. But this is a case where the load can be calculated with great precision taking the maximum

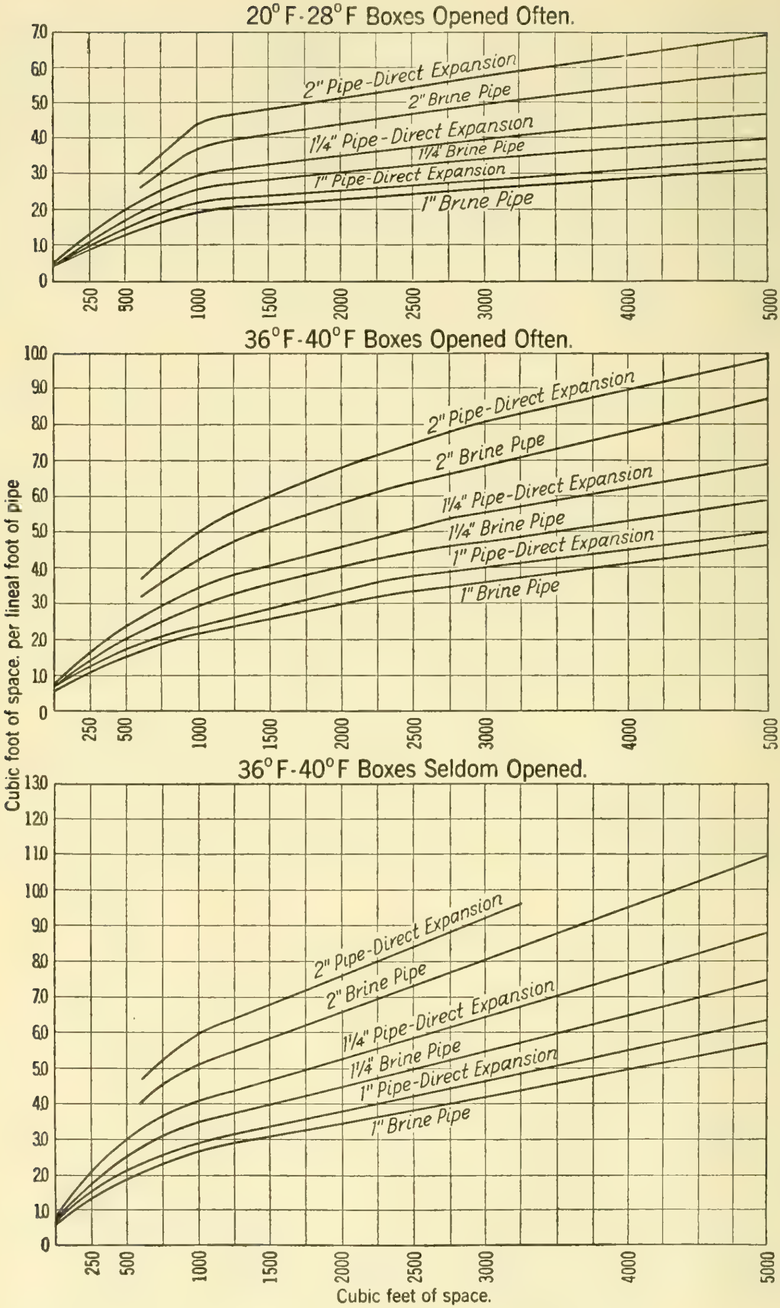


FIG. 284.—Piping for Small Cold Storage Boxes.

sustained temperature usual in the locality where the installation is to go, and designing the plant for not more than 12 hours' operation, thereby giving a factor of safety if more severe conditions prevail.

Ozone.—Ozone when used in cold storage work is in amounts less than one volume in one million, which concentration has been found to be non-germicidal. Ozone cannot be made to kill flourishing fungi when used in such small concentrations, but it is quite possible to prevent the *development* of *fungi* by its use. Ozone[13] will inhibit the development of spores but will not destroy mold growth. "Ozone is a prophylactic and not a remedial agent. It has been found by experiment that air with a mild concentration of ozone has prevented mold growth in egg storage carrying a relative humidity of 90 per cent. In egg and bacon rooms with very high humidity, molds have been effectively inhibited. Equally good results have been obtained in general cold storage practice." Mr. Hartman says that all odorous substances give off gases or finely divided particles, and when oxidized they become non-odoriferous. Ozone has been found satisfactory for freeing the atmosphere of these gases, and thus preventing taste transfer or modification. Onions can be stored in the same room with fruits and vegetables without transfer of taste, and it has been found possible to combat mold formation on grapes and other fruits. In storage of such commodities as cheese, the air in the room has been maintained fresh and free from odors, whereas ozone itself has never caused deleterious effects on the goods stored.

Calculating the Refrigerating Load.—The refrigerating load is very hard to calculate at times with accuracy, and so it has become the custom of refrigerating engineers to use the method of the rule of thumb. This is because there are so many variable factors which may affect the machine capacity or the amount of piping in the room that it has been found desirable to install the refrigeration or the piping surface from experience with other and similar jobs where satisfactory results have been obtained. For example, Fig. 284 gives a set of curves of such an origin for different sizes of boxes and operating temperatures and conditions, such as whether the boxes are to be "opened frequently" or "seldom." It is quite evident that such curves have a value, but the box that would be satisfactory in New York need not necessarily be satisfactory in Alabama. These curves then are for the practice on which they were based, and similar curves are sometimes marked for "good insulation" and for "poor insulation." However, the small cold storage box, say of 5000 cu. ft. interior contents and

[13] F. E. Hartman, American Society of Refrigerating Engineers Journal, November, 1924.

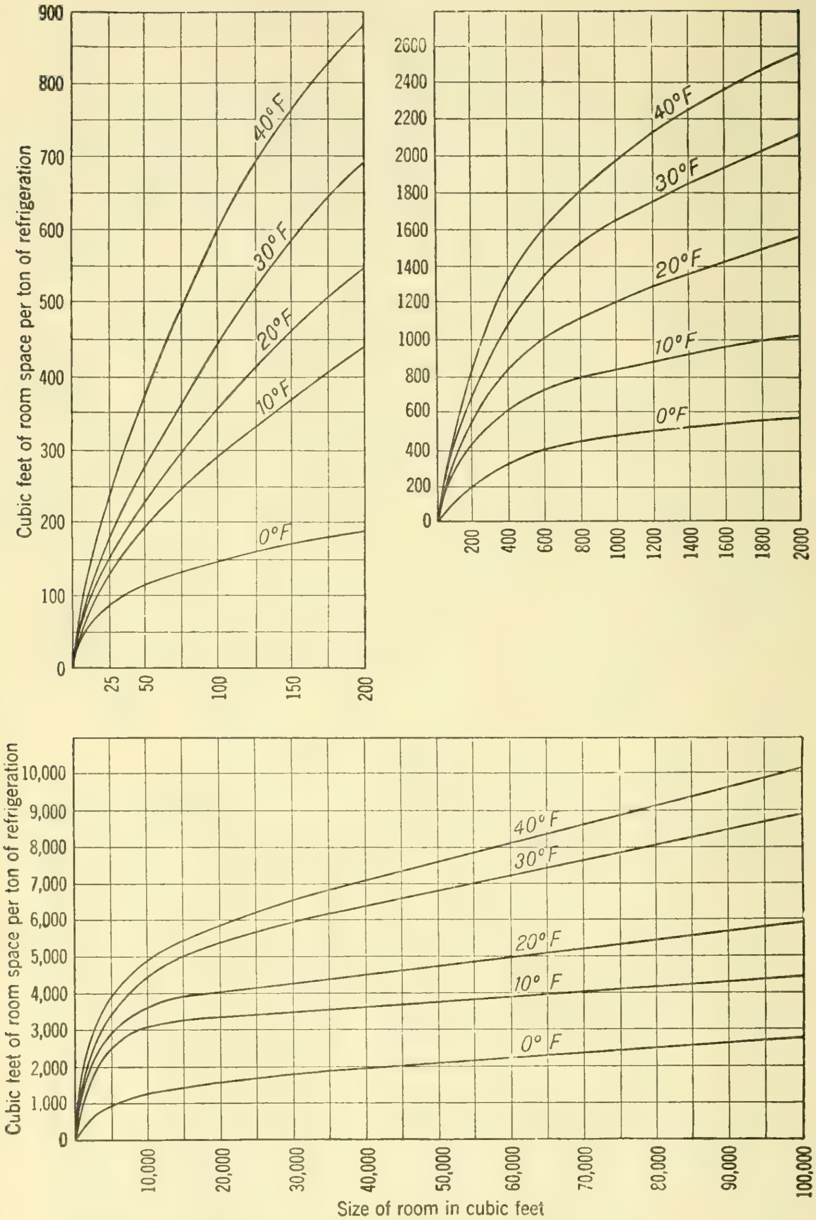


Fig. 285.—Piping for Cold Storage Rooms with Usual Insulation.

under, is practically impossible of accurate calculation as regards the load to be encountered. Such boxes are carelessly cared for, the doors are opened perniciously and often left open needlessly. The small box

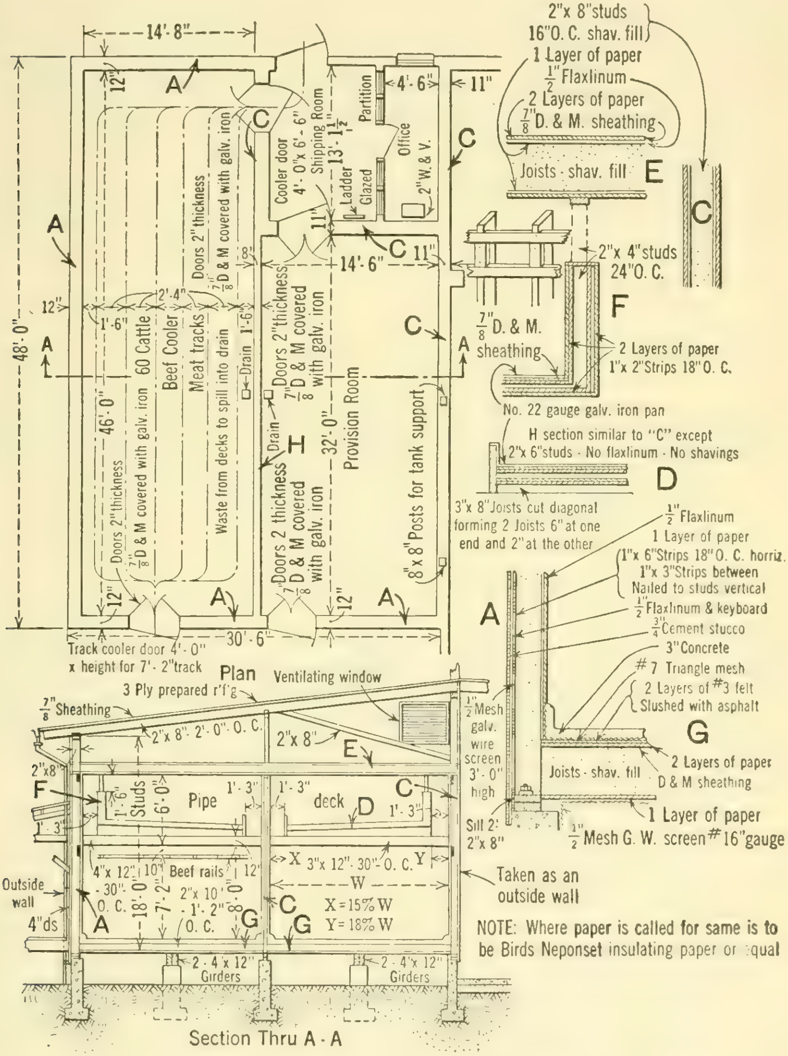


FIG. 286.—Construction of Cold Storage Rooms [for Problem].

has (unless built for some particular purpose) no well-defined load, and so the surface installed in such boxes must be liberal.

The piping for the large cold storage room (Fig. 286), on the contrary,

is more easily calculated. Such a room is likely to be of permanent building construction, and the insulation will be comparable with the rest of the building. The usual manner of calculation is to make an estimate of the heat leakage under the assumed conditions and then the refrigeration required to cool the "live load." The following illustrative example will serve as a guide, and such construction is representative of the temporary buildings constructed by the United States Government for the army cantonments. The calculation is identical with those used in Chapter VI in solving for heat leakage.

Cold Storage Problem.—The cold storage rooms are shown in Fig. 286. The live load will consist of the following commodities received daily:

1 car of beef (a car holds 40 beeves at 600 lb.).
20 barrels of apples.
50 barrels of potatoes.
20 cases of eggs (of 60 lb.).
10 tubs of butter.

The beef and butter arrive at 60 deg. F. The apples, potatoes and eggs will be taken as arriving at 80 deg. F. The operating suction pressure will be taken at 30 lb. gage.

Take the specific heat of beef = 0.77 (average)
apples = 0.92
potatoes = 0.80
butter = 0.60
eggs = 0.76

The outside temperature will be taken at 90 deg. F., and the temperature in the beef and the provision rooms will be held at 33 degrees. The first problem will be the calculation of the heat leakage.

Wall "A".—This is composed of $\frac{7}{8}$-in. sheathing, $\frac{1}{2}$-in. flaxlinum, 9 in. of shavings, $\frac{3}{4}$ in. stucco, and $\frac{1}{2}$ in. flaxlinum (neglect insulation effect of building paper).

Using the formula for heat leakage

$$u = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2} + \frac{T_1}{C_1} + \frac{T_2}{C_2} + \frac{T_3}{C_3} + \frac{T_4}{C_4}, \text{ etc.}}.$$

using the values for this construction,

$$u = \frac{1}{\frac{1}{1.4} + \frac{1}{4.2} + \frac{1}{1.0} + \frac{(\frac{1}{2} + \frac{1}{2}) \times 24}{7.9} + \frac{9 \times 24}{10} + \frac{0.75}{8.0}} = 0.03745.$$

Ceiling "E."—2-$\frac{7}{8}$ in. sheathing, 2–$\frac{1}{2}$ in. flaxlinum, and 8 in. shavings.

$$u = \frac{1}{\frac{1}{1.4} + \frac{1}{4.2} + \frac{2}{1} + \frac{(\frac{1}{2} + \frac{1}{2}) \times 24}{7.9} + \frac{8 \times 24}{10}} = 0.0397.$$

Wall "C."—This is the same as the E construction.

$u = 0.0397.$

Floor "G."

$$u = \cfrac{1}{\cfrac{1}{0.93} + \cfrac{1}{4.2} + \cfrac{3}{1} + \cfrac{10 \times 24}{10} + \cfrac{3.0}{8.0}} = 0.03486.$$

Heat Leakage (meat room):

Sides and 2 ends	$0.03745 \times (644 + 205 + 205)(90 - 33)$	= 2250 B.t.u.
Floor	$0.03486 \times 674 \times (90 - 33)$	= 1340 B.t.u.
Ceiling	$0.0397 \times 674 \times (90 - 33)$	= 1526 B.t.u.
Side	$0.0397 \times 448 \times (33 - 33)$	= 0 B.t.u.
Side (shipping room)	$0.0397 \times 184 \times (90 - 33)$	= 416 B.t.u.
	Total	= 5532 B.t.u.
	Add 20 per cent	1106
		6638 B.t.u. per hr.

Heat Leakage (provision room):

Side and one end	$0.0397 \times (448 + 203)(90 - 33)$	= 1474 B.t.u.
Floor	$0.03486 \times (464)(90 - 33)$	= 922 B.t.u.
Ceiling	$0.0397 \times 464 \times (90 - 33)$	= 1050 B.t.u.
One end	$0.03745 \times 203 \times (90 - 33)$	= 433 B.t.u.
	Total	= 3879 B.t.u.
	Add 20 per cent	776 B.t.u.
		4655 B.t.u. per hr.

The second problem is the calculation of the live load. The assumption is that the temperature of the commodities is reduced to 33 deg. F. in 24 hours.

Live Load	Specific Heat	Degrees Cooling	Total Weight	Total Refrigeration, B.t.u.
Beef	0.77	27	40×600	499,000
Apples	0.92	47	20×150	129,500
Potatoes	0.80	47	50×180	338,500
Butter	0.60	27	10×63	10,200
Eggs	0.76	47	20×60	42,900
Total				521,100

Total Load, meat room, per hour $= 499{,}000 \div 24 + 6638 = 27{,}430$ B.t.u. per hour.
Provision room $= 521{,}100 \div 24 + 4655 = 26{,}360$ B.t.u. per hour.

Piping required:

Meat room	$27{,}430 \div [1.6 \times (33.0 - 16.6)]$	= 1045 sq. ft.
		= 2405 lin. ft. $1\frac{1}{4}$-in. pipe.
Provision room	$26{,}360 \div [1.6 \times (33.0 - 16.6)]$	= 1002 sq. ft.
		= 2320 lin. ft. $1\frac{1}{4}$-in. pipe.

The arrangement of the piping will be as indicated by Fig. 287, which shows the expansion valve between two stop valves and the liquid feed at the bottom of the coils. An allowance of 20 per cent excess heat leakage is made for the opening of doors, the heating effects of lights and of workmen in the room, etc. It is assumed that the live load will be reduced to such an extent that the temperature of the commodities will be taken at 33 deg. F. at the end of 24 hours, and that the coefficient

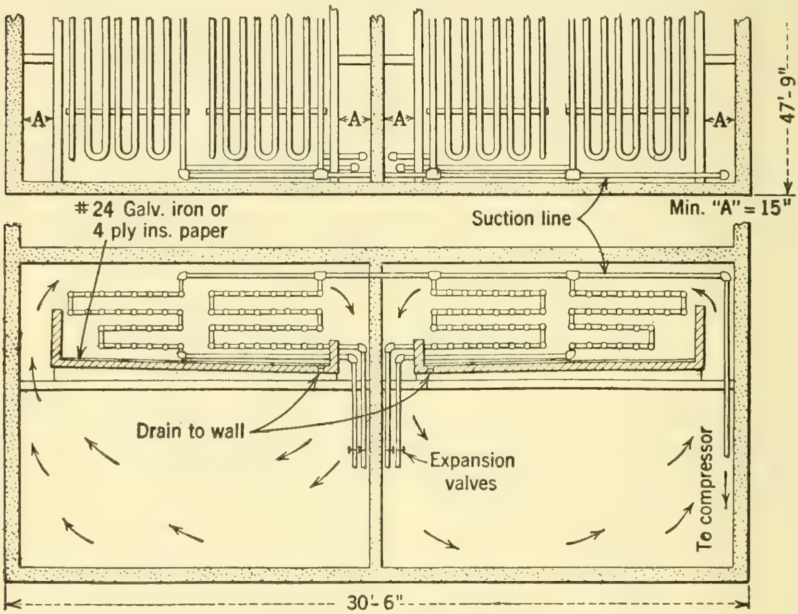


Fig. 287.

of heat transfer (k) of the pipes is equal to 1.6 B.t.u.[14] The tonnage on the machine becomes $\dfrac{27{,}430 + 26{,}360}{12{,}000} = 4.47$ tons. The size of the suction line is found (at 155 lb. condenser pressure and 30 lb. suction pressure, and with a velocity of the gas in the return suction line of 4000 ft. per minute, by the following method): Volume of gas $\dfrac{200}{616.8 - 139.1} \times 6.35 \times 4.47 = 11.9$ cu. ft. per min. The cross-

[14] The value of 1.6 for k is low, but is conservative on account of possible frostation of the pipes and careless operation.

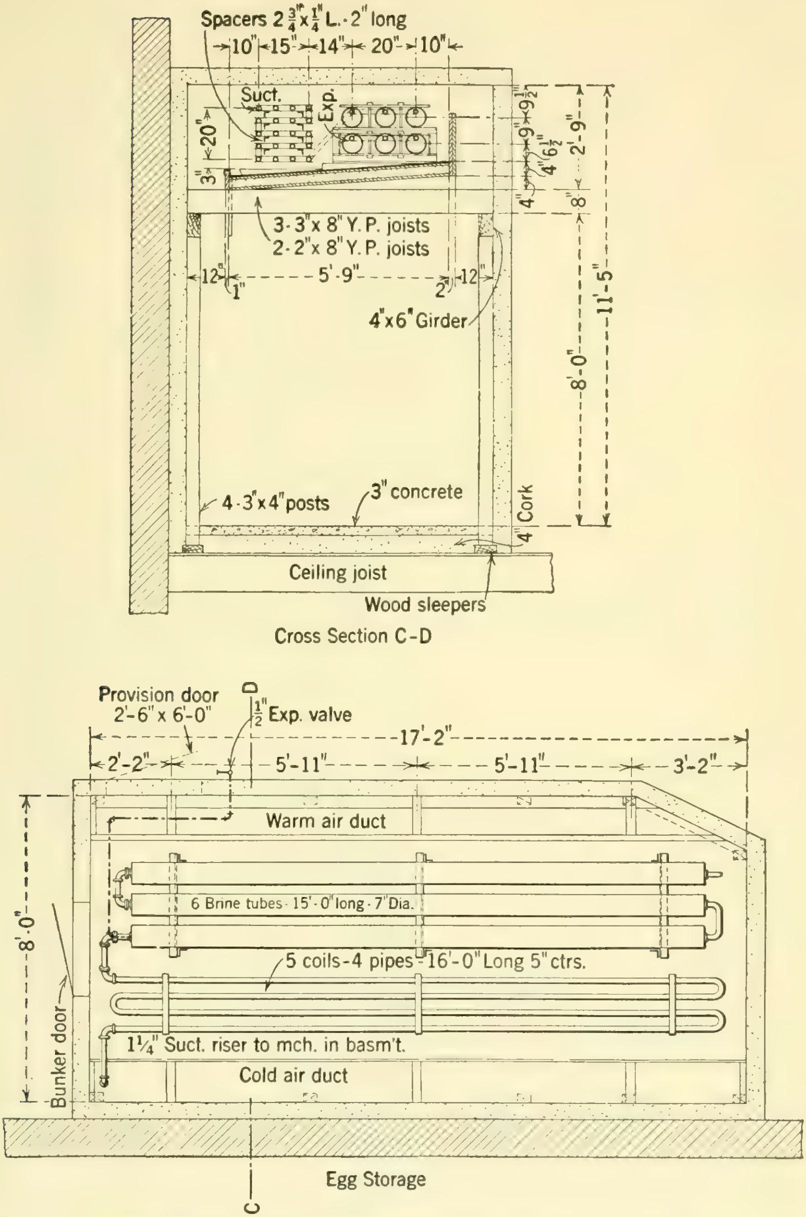


FIG. 288.—Brine Tubes and Ammonia Piping in Pipe Deck.

TABLE 96

COMPOSITION AND SPECIFIC HEAT OF FOOD PRODUCTS AND STORAGE TEMPERATURES

Product	Temperature Carried	Composition, Per Cent		Specific Heat above Freezing	Specific Heat below Freezing	Latent Heat of Freezing
		Water	Solids			
Lean beef	30	72.00	28.00	.77	0.41	104
Fat pork	30	39.00	61.00	.51	.30	56
Eggs	30	70.00	30.00	.76	.40	101
Potatoes	34	74.00	26.00	.80	.42	107
Cabbage	33	91.00	9.00	.93	.48	131
Carrots	33	83.00	17.00	.87	.45	120
Cream	33	59.25	30.75	.68	.38	85
Milk	35	87.50	12.50	.90	.47	126
Oysters	35	80.38	19.62	.84	.44	116
Whitefish	15	78.00	22.00	.82	.43	112
Chickens	28	73.70	26.30	.80	.42	106
Ice	23	100.00	0.00	1.00	.504	144

APPROXIMATE COLD STORAGE TEMPERATURES

Article	Temperature, Degrees F.	Article	Temperature, Degrees F.
Apples	30	Huckleberries, frozen	20
Asparagus	33	Ice	28
Bananas	55	Ice cream, short carry	15
Beans, fresh	32	Lemons, short carry	50
Beans, dried	45	Lemons, long carry	38
Beef, fresh, short carry	35	Lambs	32
Beef, fresh, long carry	30	Lard	40
Beef, dried	40	Livers	20
Beer, in barrels	32	Maple syrup and sugar	45
Beer, in bottles	45	Meats canned	40
Berries, fresh, short carry	40	Meats, salt, after curing	43
Buckwheat flour	42	Milk, short carry	35
Butter	14	Nursery stock	30
Butterine	20	Nuts in shell	40
Cabbage	33	Oatmeal	42
Cantaloups, short carry	40	Oils	45
Cantaloups, long carry	33	Oleomargarine	20
Carrots	33	Onions	32
Celery	32	Oranges, short carry	50
Cheese, long carry	35	Oranges, long carry	34
Chestnuts	34	Oxtails	30
Chocolate dipping room	65	Oysters in shell	43
Cider	32	Oysters in tubs	35
Cigars	42	Parsnips	32
Corn, dried	45	Peaches, short carry	50
Cornmeal	42	Pears	33
Cranberries	33	Peas, dried	45
Cream, short carry	33	Plums	32
Cucumbers	38	Potatoes	34
Currants, short carry	32	Poultry, dressed, iced	30
Dates	55	Poultry, short carry	28
Eggs	30	Poultry, after frozen	10
Figs	55	Poultry to freeze	0
Fish, not frozen, short carry	28	Raisins	55
Fish, fresh water, frozen	18	Ribs, not brined	20
Fish, salt water, not frozen	15	Salt meat curing room	32
Fish, to freeze	5	Sardines, canned	40
Fish, dried	40	Sauerkraut	38
Flowers, cut	36	Sausage casings	20
Fruits, canned	40	Scallops, after frozen	16
Fruits, dried	40	Sheep	32
Furs	28	Shoulders, not brined	20
Game, short carry	28	Sugar	45
Game, after frozen	10	Syrup	45
Game, to freeze	0	Tenderloins	33
Ginger ale	36	Tobacco	42
Grapes	36	Tomatoes, ripe	42
Hams, not brined	20	Watermelons, short carry	40
Hogs	30	Wheat flour	42
Honey	45	Wines	50
Hops	32	Woolens	28

TABLE 96—*Continued*

COLD STORAGE PRACTICE

Temperature of Storage, Degrees F.	Humidity, Per Cent	Allowances
		1 bbl. = 31½ gal. of 231 cu. in. each.
		1 bushel = 1.245 cu. ft. = 2150.4 cu. in.
35	80 to 90	1 bbl. potatoes = 5 cu. ft. = 180 lb. = 2⅓ bushels.
32 to 34	70 to 80	1 box cheese = 60 lb. = 2 cu. ft.
30 to 31	75	1 case eggs = 30 doz. = 50 – 70 lb. = 2¼ cu. ft. = 12 × 13 × 25.
2 or lower	70	1 tub butter = 63 lb. = 1½ or 2 cu. ft. (piles not over 6 ft.).
30 to 35	85	1 bbl. apples = 150 lb., 5 cu. ft. = 2½ bushels.
34	80 to 90	1 crate celery = 10 cu. ft. = 140 lb. = 24 × 24 × 30 in. (8 ft. high).
34	80 to 90	1 bbl. vegetables = 5 cu. ft., piles 5 ft. high.
35	80	1 box oranges (packed on end) = 4 cu. ft., 70 lb. = boxes 15 × 15 × 30.

Aisles—Allow ⅓ for eggs, celery and oranges.
Allow ⅕ for potatoes and vegetables.
Allow ¼ for mutton, pork, poultry (4 ft. high).
Specific gravity of boned beef = 1073 = 67.2 lb. per cu. ft.

sectional area of the suction pipe $= \frac{11.9 \times 144}{4000} = 0.428$ sq. in., and as the ¾-in. pipe has a transverse area of 0.533 and the ½-in. pipe has one of 0.304 sq. in. the ¾-in. pipe would be the one to be chosen for satisfactory results unless the connections on the compressor are for a larger pipe and it seems desirable to make the entire suction pipe of this size.

Hold Over Tanks and Pipes.—Small plants find it uneconomical to operate at times more than 10 to 12 hours per day. Such plants, if considerable temperature rise is to be prevented, require under these conditions either brine operation and a brine tank of sufficient storage capacity to carry the load during periods of shut-down of the compressor, or must provide some auxiliary means to absorb the heat leakage during this part of the day. The accepted method is to use brine storage (of congealing) tank or tubes. In either case a certain amount of brine is cooled during the normal operation of the compressor, at the same time that the room is cooled by the usual piping, and piping arrangements, by means of direct expansion piping submerged by the brine, are so located as to take the place of the direct expansion piping. This scheme is seldom better than a compromise, but there is some advantage in its use. At times 6-in., 7-in. or 8-in. pipe can be used to better advantage than a tank, in which case the pipes are filled with brine. If in the previous problem the provision room was to be controlled during 12 hours by the brine pipe the leakage would be (for 70 deg. F. average temperature outside) 2520 B.t.u. per hour, and in 12 hours this would accumulate to 30,240 B.t.u. If the brine is cooled 20 degrees below the temperature of

the room (that is to 13 deg. F.), and the brine rises 15 degrees during the 12 hours, the requred amount of brine becomes

$$30{,}240 = \text{Weight} \times 0.83 \times 15.$$

$$\text{Weight of brine} = 30{,}240 \div 12.45 = 2430 \text{ lb.},$$

and using a specific gravity of brine of 1.16,

$$2430 \div (1.16 \times 62.3) = 33.7 \text{ cu. ft.}$$

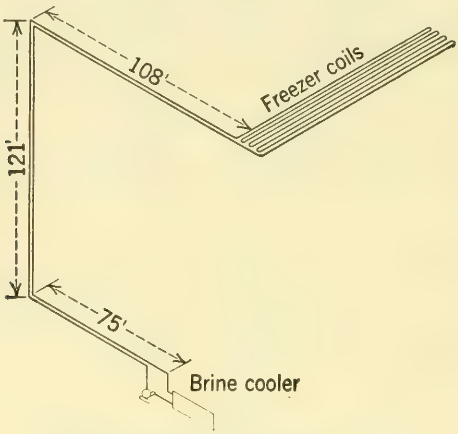


FIG. 289.—Problem in Brine Piping.

TABLE 97

REFRIGERATED SPACE—SUMMARY BY CLASS OF BUSINESS, OCTOBER 1, 1925

Class of Business	Number of Concerns	Space, in Cubic Feet, Held at Temperatures of 10 Deg. F. and below	11 Deg. to 29 Deg. F. inclusive	30 Deg. to 44 Deg. F. inclusive	45 Deg. F. and above	Total Space
Public cold storage	416	41,311,130	20,196,430	175,244,429	5,811,924	242,563,913
Private cold storage	264	3,288,072	5,313,301	14,291,508	1,172,946	24,065,827
Combined public and private cold storage	212	8,450,311	11,173,425	36,383,333	1,027,711	57,034,780
Meat-packing establishments	397	11,530,051	16,458,902	195,026,108	23,150,561	246,165,622
Meat-packing establishments doing a public cold-storage business	34	4,142,218	6,272,375	41,067,020	5,355,995	56,837,608
Total refrigerated space	1323	68,721,782	59,414,433	462,012,398	36,519,137	626,667,750

The number of pipes for this amount of hold-over brine can be calculated when the other details are known. If 8-in. pipe is used, then 2.88 ft. are required to contain one cubic foot, and the total linear feet required will be 97.1.

Friction Head in Brine Pipes.—Considering the importance of brine in refrigeration it is surprising how little information on the friction head

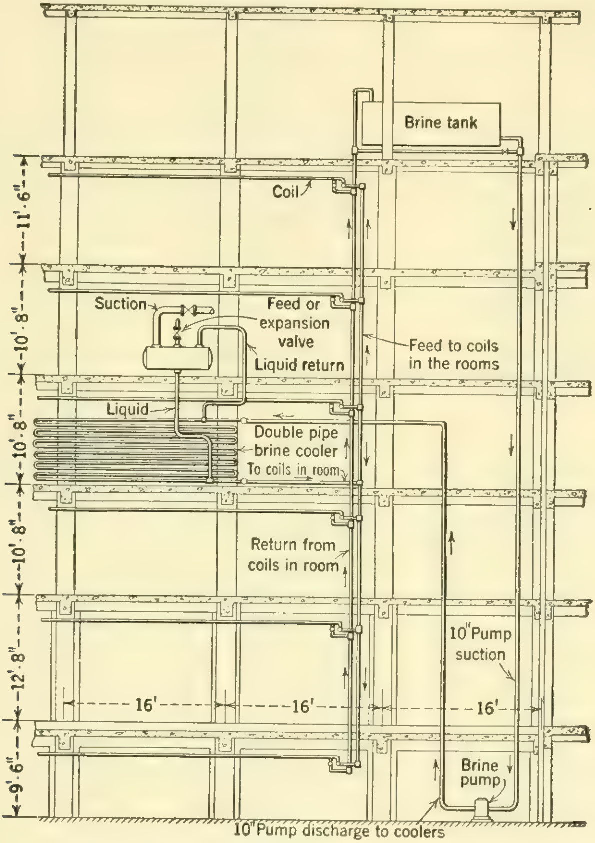


FIG. 290.—Typical Installations—Brine.

loss due to the flow of brine in pipes has so far been collected. The only information on the subject is that of A. H. Gibson, of the University College, Dundee,[15] in which it is shown that the coefficient of friction head increases both with the specific gravity and with a decrease of temperature. The results of his experiments are shown in Fig. 181,

[15] Professor A. H. Gibson, Proceedings of the Inst. of Mech. Eng., London, Feb., 1914. See Chapter VIII for details on this paper.

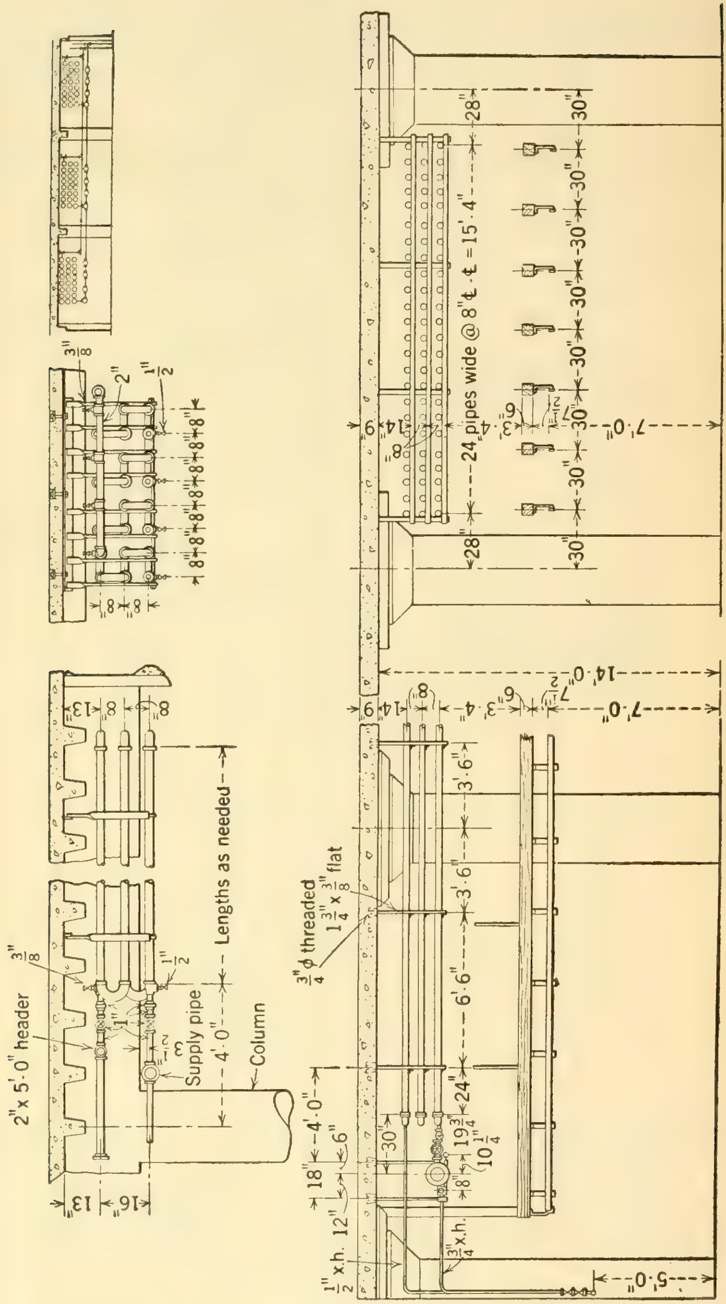


FIG. 291.—Typical Installations—Ammonia.

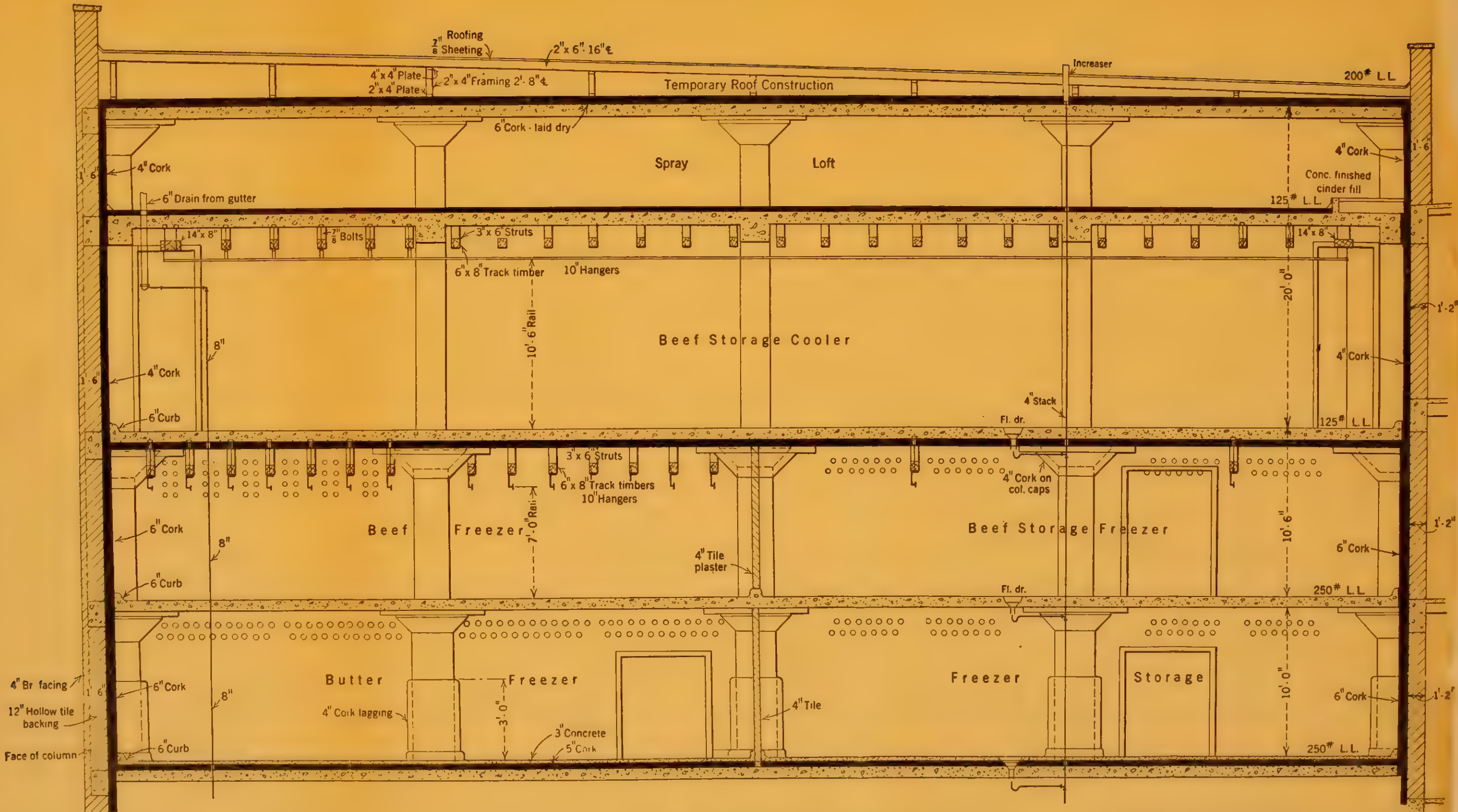


Fig. 292.—Typical Installations—Packing House.

To face page 46

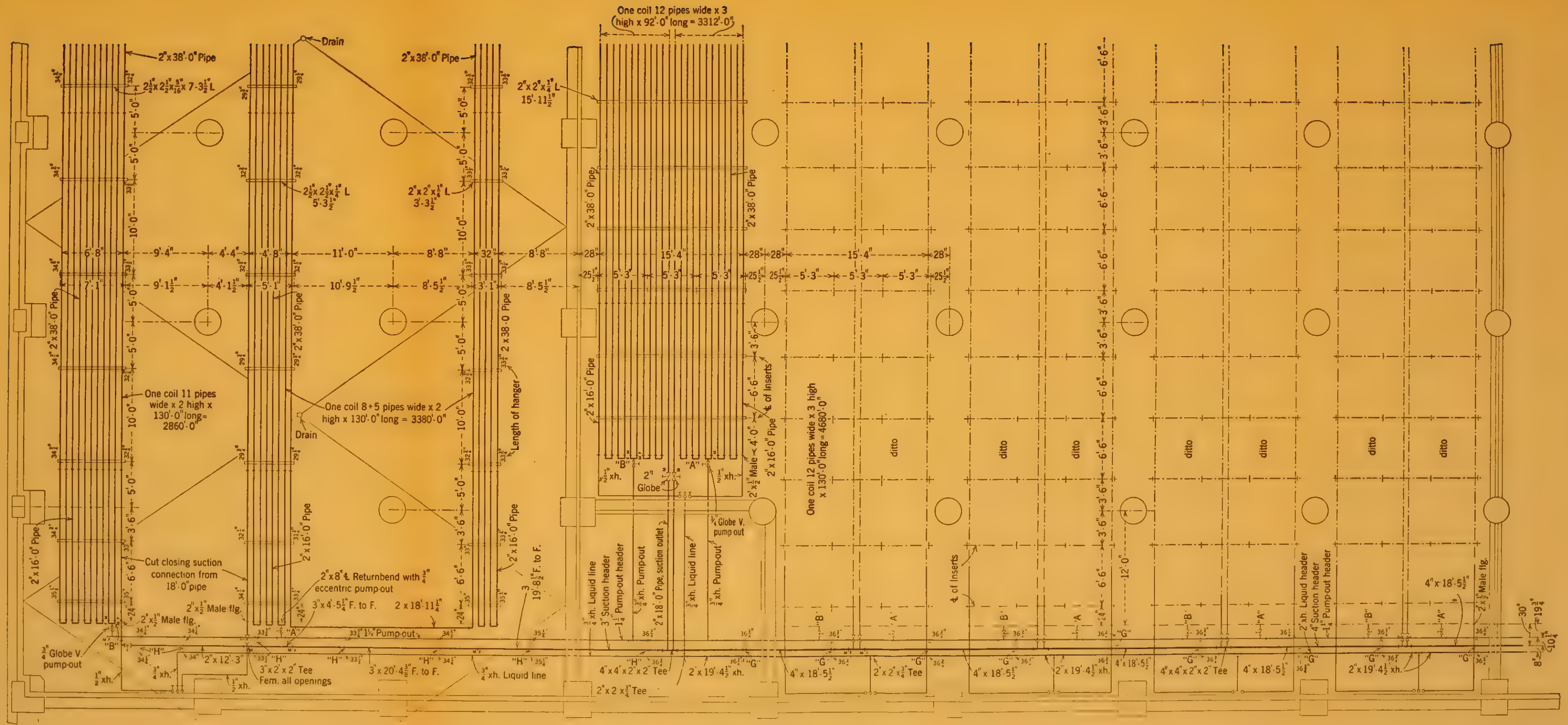


FIG. 293.—Typical Piping—Ammonia.

To face page 463.

which give *multiplying factors* (with reference to the friction head of water) to be used in conjunction with Fig. 180. To use these figures the friction head will be found as if the fluid flowing were water and the proper multiplying factor must be found from Fig. 181. The use of the tables will be shown in the following problem.

Problem.—A brine coil of 10,000 lin. ft. of 2-in. pipe is located on the 10th floor of a warehouse. The brine circuit consists of the following: The refrigerating coils, of 2-in. pipe, 10,000 lin. ft. in 10 coils; distance from the brine cooling coils to riser, 108 lin. ft. 3-in. pipe; the riser, 121 lin. ft. 3-in. pipe; riser to brine cooler and pump, 75 lin. ft. 3-in. pipe; total pipe in the circuit, 10 coils of 2 in. each 1000 ft. and 304 ft. of 3-in. pipe.

The freezer room is to be maintained at 32 deg. F. and the brine will enter at 18 degrees and will leave at 22 degrees. The coefficient of heat transfer will be taken at 1.7 B.t.u. for the piping. The amount of brine to be circulated, the friction head loss and the horse power of the pump assuming an overall efficiency of 50 per cent is desired. The arrangement of the piping is indicated in Fig. 290.

The amount of brine to be circulated may be found by the amount of cooling surface. Using a value of $k = 1.7$ B.t.u., the total refrigeration becomes $(10{,}000 \div 1.6) \times 1.7 \times \left(32 - \frac{22 + 18}{2}\right) = 127{,}400$ B.t.u. per hour where the factor 1.6 is the number of feet of 2-in. pipe per one sq. ft. of outside surface.

$$127{,}400 = \text{wt. brine} \times 0.76 \times 4.$$

$$\begin{aligned}\text{Wt.} &= 127{,}400 \div (0.76 \times 4) = 41{,}900 \text{ lb.}\\ &= 41{,}900 \div (62.4 \times 1.15) = 584 \text{ cu. ft. per hour}\\ &= 72.7 \text{ gal. per minute.}\end{aligned}$$

From the chart it will be seen that at a velocity of 3.4 ft. per sec. a 3-in. pipe will have a pressure drop per 100 ft. of pipe of 0.72 lb. (for water) and the friction head loss in the cooling coils (7.3 gal. per min. in 2-in. pipe) will be 0.072 lb. per 100 ft. The total friction head loss is then,

$$0.072 \times 1.3 \times 10 = 0.935 \text{ lb.}$$

$$\frac{(2 \times 108 + 2 \times 121 + 2 \times 75)}{100} \times 0.72 \times 1.3 = 5.69$$

$$\begin{aligned}\text{Total} &= 6.63 \text{ lb.}\\ \text{Add 20 per cent for fittings, etc.} &= 1.33\\ \text{Total} &= 8.0 \text{ lb.}\\ \text{Work performed} = (41{,}900 \div 60) \times 16.0 &= 11{,}200 \text{ ft.-lb.}\\ 11{,}200 \div 33{,}000 &= 0.340 \text{ hp. for circulating the brine.}\\ &= 0.680 \text{ hp. of the pump assuming 50 per cent efficiency.}\end{aligned}$$

This work performed is under the conditions of a balanced system for the brine. Certain types and designs are shown in Figs. 290 to 293. When the open brine design is used with sprays, the head on the pump is the actual lift plus the friction and the velocity heads. The work performed is then much greater.

THE PACKING HOUSE

The Extent of the Industry.—According to the Bureau of Agriculture Economics for October, 1921, the total refrigerating space in the packing plants in the United States amounts to 258,549,000 cu. ft. Of the amount, which includes 443 plants, 15,400,000 cu. ft. are held at 10 deg. F. or lower; 18,300,000 are held at temperatures between 11 and 29 deg. F., and 200,000,000 cu. ft. are held at temperatures ranging from 30 to 44 deg. F. The total refrigerating space, according to the same authority, is 543,600,000 cu. ft. Of this, Armour and Company have 17,000,000 cu. ft. in Chicago, and 9,500,000 cu. ft. in Kansas City, requiring 4380 and 3120 tons of refrigeration respectively; Morris and Company have 13,000,000 cu. ft. in Chicago and 8,000,000 cu. ft. in Kansas City, requiring 3450 and 2100 tons of refrigeration respectively.

Published statistics by the Bureau of Animal Husbandry of the United States Department of Agriculture, 1921, indicate that a total of 63,415,000 animals were slaughtered in 1920 under inspection, including 7,600,000 cattle, 3,800,000 calves, 13,000,000 sheep and lambs, and 39,000,000 hogs. At least 80 per cent of this total amount requires mechanical refrigeration for a short time at least, although the average time is between one and two months, and the average time after slaughtering before the meat is consumed is less than two months. A very small percentage (5 per cent) of the meat is frozen and is held in freezer storage.

The packing house practice has been a development, as in other applications of refrigeration. There are two main operations—the chilling of the hot freshly slaughtered animals and the maintaining of cold or freezer storage. In general "forced" draft or the forcing of chilled air through pipes or ducts is not used in American practice, because of the belief that it causes shrinkage, tends to darken the meat, that the increased pressure necessary to circulate the air causes excessive air leaks, and because of the expense for the operation of the fan and the space occupied by the air ducts. A modified system using "sprays" either in a loft, troughs, or in pipes is a modern successor to the old duct system.

The use of direct expansion or of brine is a matter for individual decision, although certain factors tend to swing the preference towards the use of brine. One of the important reasons is because of the danger in the use of direct expansion of ammonia by the class of labor to be found in the packing plant. Also the long pipe systems represent a heavy investment of ammonia—a large part of which would be lost (with a

TABLE 98

Suction pressure		45	40	35	30	27	25	22	19	15	12	10	7	4.7	1.4	−2.4
Temperature of NH_3, deg. F.		30	26	22	17	14	12	8	4	0	−5	−8	−13	−18	−25	−35
Temperature of brine, deg. F.		38	34	30	25	22	20	16	12	8	3	0	−5	−10	−19	−30
Room Temperature, Degrees F.	Piping															
65	2-in. Dr. Exp.	220	198	179	160	151	145	135	126	119	110	106	99	93	86	77
	1¼-in. Dr. Exp.	314	282	256	229	216	208	193	180	170	157	151	141	133	123	110
	1¼-in. Brine	400	349	309	270	252	240	221	204	190	178	167	155	144	129	114
60	2-in. D. E.	257	227	203	179	168	161	148	138	129	119	114	106	99	91	81
	1½-in. D. E.	368	324	290	256	239	229	212	197	184	170	162	151	141	130	116
	1¼-in. Br.	491	416	360	309	285	270	246	225	208	190	180	167	155	137	120
55	2-in. D. E.	308	266	234	203	188	179	164	151	140	129	122	113	106	96	86
	1¼-in. D. E.	440	380	335	290	269	256	244	216	200	184	175	163	152	137	123
	1¼-in. Br.	635	515	433	360	328	309	277	251	230	208	197	180	167	146	127
50	2-in. D. E.	365	321	275	234	214	203	184	168	154	140	133	122	114	103	91
	1¼-in. D. E.	550	459	393	337	306	290	264	240	220	200	193	175	163	147	130
	1¼-in. Br.	900	675	540	432	386	360	318	284	258	230	216	197	180	157	135
45	2-in. D. E.	514	406	335	275	249	234	208	188	171	154	145	133	123	110	96
	1¼-in. D. E.	735	580	478	393	356	334	297	269	244	220	207	190	176	157	137
	1¼-in. Br.	1540	982	720	540	470	432	373	327	292	258	240	216	197	169	144
40	2-in. D. E.	770	550	428	335	297	275	241	214	193	171	161	145	133	119	103
	1¼-in. D. E.	1110	785	612	478	425	393	344	306	276	244	232	207	190	170	147
	1¼-in. Br.	5400	1800	1080	780	600	540	450	386	338	292	270	240	216	184	154
35	2-in. D. E.	1540	856	592	427	366	334	286	248	220	192	179	160	145	129	110
	1¼-in. D. E.	2220	1220	845	610	523	477	409	355	314	274	256	229	207	184	157
	1¼-in. Br.	......	......	2160	1080	830	720	569	470	400	337	308	270	240	200	166
30	2-in. D. E.	......	1925	962	592	481	427	350	297	256	220	202	179	160	140	119
	1¼-in. D. E.	......	......	......	845	688	610	500	424	366	314	288	256	229	200	170
	1¼-in. Br.	......	......	......	2160	1350	1080	772	600	490	400	360	308	270	220	180
25	2-in. D. E.	......	......	2566	962	700	592	452	367	308	256	233	202	179	154	129
	1¼-in. D. E.	......	......	......	1375	1001	846	645	525	440	366	333	288	256	220	185
	1¼-in. Br.	......	......	......	......	......	2160	1200	830	636	490	432	360	308	246	196
20	2-in. D. E.	......	......	......	2566	1283	962	642	482	385	308	275	233	202	171	140
	1¼-in. D. E.	......	......	......	......	......	1375	917	689	550	440	393	333	289	244	200
	1¼-in. Br.	......	......	......	......	......	......	2700	1350	900	636	540	432	360	277	216
15	2-in. D. E.	......	......	......	......	7700	2566	1100	700	514	385	334	275	233	193	154
	1¼-in. D. E.	......	......	......	......	......	......	......	1000	734	550	477	293	333	276	220
	1¼-in. Br.	......	......	......	......	......	......	......	3600	1543	900	520	540	432	318	240
0	2-in. D. E.	......	......	......	......	......	......	......	......	7700	1540	962	592	527	308	220
	1¼-in. D. E.	......	......	......	......	......	......	......	......	......	......	1375	846	754	441	315
	1¼-in. Br.	......	......	......	......	......	......	......	......	......	......	......	2160	1080	569	360
−10	2-in. D. E.	......	......	......	......	......	......	......	......	......	......	......	2570	960	514	308
	1¼-in. D. E.	......	......	......	......	......	......	......	......	......	......	......	......	1375	735	441
	1¼-in. Br.	......	......	......	......	......	......	......	......	......	......				1200	540

Factor "K" = B.t.u. transmitted per square foot of pipe per 1 deg. difference per hour.

Table is based on heat transmission of K = 2.5; if any other value of K is desired multiply by 2.5 and divide by new value of K. If no frost accumulates on pipe, multiply lineal foot by 0.6. For forced air circulation multiply by .5 to .6.

forced shut-down) by accident to any part of the ammonia system, as well as likelihood of heavy replacement expense due to flange or other leaks. Brine is much more flexible, the temperature is easy to control, " off peaks " operation is possible, brine spraying is available and the piping is cheaper. When necessary, a − 20 deg. F. calcium chloride brine can be carried.

The Balanced System.—When brine is used the preferred method is to use a balanced system whenever possible. By this means the piping acts as a syphon and the pumping cost is that of overcoming the pipe friction only. By the proper use of the balance tank, the pipes are kept full of liquid all the time, and the pressure against which the pump has to work should be only 20 to 25 lb. for a nine story building. This is called a *closed* brine system. The cooler lofts sometimes have brine defrosting pipes, in which case some form of concentrator should be provided in order to conserve the salt. One method of removing the frost is to arrange a slotted 2-in. brine pipe over each bank, so piped that defrosting is possible by permitting brine to drop from pipe to pipe. Usually one-third of the coils are defrosted at a time. Heavy frost accumulations are very detrimental to heat transfer, and the refrigerating ability is thereby reduced. At times one hears that as the outside of the pipe becomes larger the total transfer of heat must remain practically constant. The same argument would be true in the case of pipe covered with magnesia or corkboard (as far as the outside diameter of the surface is concerned). The brine flowing over the pipes must be kept concentrated so as not to *freeze* on the pipes.

If an " open " brine system is used there is an advantage at times in installing the balance tank at a point best located to receive the drain from the brine spray nozzles. The closed pipe portion also empties into the tank and a booster pump supplies the extra pressure required for the sprays. Any open brine system will absorb moisture from the air and in the best chill rooms the shrinkage is at least $1\frac{1}{2}$ to 2 per cent. As the steam from the carcasses must be absorbed by the brine it is evident that the brine must be either thrown away or reconcentrated. One type of concentrator is a Baudalot arrangement where the brine passes *upward* inside 17 pipes, then up to a trough at the top of the stand and then downwards on the outside of six steam-heated pipes and then outside the 17 pipes already mentioned as a counterflow to the brine inside of the pipes. The six upper pipes provide the heating surface for the concentration, although it is evident that the next two or three are of some assistance in providing the time required for the removal of the steam. An open vat (steam heated) could be used, but if the exchanger principle is lost the design is not an economical one.

TABLE 99

Wholesale Meat Markets

Refrigeration Data

	Horse-power	Tons	Cubic Feet	Kw. Hours per Year	Kw. Hours per Year per Cubic Foot	Cooler Temperature, Degrees F.
1. Philadelphia, Pa.	15	8	22,980	25,728	1.12	
2. Philadelphia, Pa.	10	6	15,732	30,448	1.93	
3. Philadelphia, Pa.	25	15	21,384	42,636	2.00	
4. Chicago, Ill.	30	12½	23,190	30,456	1.31	20
5. Chicago, Ill.	10	7	7,690	14,880	1.93	20
6. Pittsburgh, Pa.	50	20	94,000	159,250	1.69	30
Packers						
7. Chicago, Ill.	30	15	4,340	6,000	1.38	36
8. Pittsburgh, Pa.	30	15	39,680	76,292	1.93	36
9. Pittsburgh, Pa.	30	15	37,450	171,144	4.61	40 (2 comp.)
10. New Orleans, La.	30	15	54,500	51,372	0.944	35
	5	2	1,440	7,580	5.27	40
	65	20	94,000	139,225	1.47	33
Provisions						
11. New York, N. Y.	15	8	10,505	23,412	2.24	36
12. New York, N. Y.	10	4	38,880	25,783	0.665	38 (makes ice)
13. Chicago, Ill.	15	8	8,820	12,204	1.38	41
Retail Meat						
14. Philadelphia, Pa.	15	8	3,000	9,949	3.3	35
15. New Orleans, La.	7	2	1,490	4,600	3.1	
16. Chicago, Ill.	15	8	4,629	26,808	5.8	36
17. Chicago, Ill.	5	2	1,706	3,408	2.0	43
18. Philadelphia, Pa.	7½	4	2,500	11,099	4.45	
Fur Storage						
19. Brooklyn, N. Y.	39¼	20	56,775	19,760	0.35	30
20. New York, N. Y.	7½	4	5,722	1,931	0.34	20
21. New York, N. Y.	15	10	33,120	23,000	0.70	16
22. New York, N. Y.	30	15	40,500	27,321	0.67	32

Chilling.—The best practice at present is to place the beef, sheep, and veal immediately in the chill rooms after the hides have been removed. The maximum time interval for standing in the air is a half hour. This chill room or fore-cooler is at 45 to 48 deg. F., and an attempt is made to bring the temperature down to 38 degrees in 12 hours. The feeling prevails in Chicago that hogs can be air dried over night and are improved in the process but careless handling may cause sour meat. The use of fore-coolers to separate freshly killed carcasses from chilled ones is still a disputed practice, but it is favored by many if space will permit it. The meat should hang 12 in. apart in the chill rooms, but heavy cattle should be 18 in. apart in order to provide circulation. The fore-cooler, if used, should be kept as near 38 deg. F. as possible, although 43 degrees is not bad for beef. If the main cooler is fitted for a fore-cooler it should be held at a maximum temperature of 38 deg. F. and it should be cooled to 34 deg. F. in 12 to 15 hours, and this 34 degrees should be held. Heavy cattle and frozen meat for export have special treatment. For example, heavy cattle should be carried at 32 deg. F. for 24 hours, then at 34 deg. F. if used for domestic trade. For export, reduce to 29 degrees as quickly as possible, and keep at that temperature until " floated." Mutton and veal are handled about the same as beef. Pork is air cooled (for the first two hours at least), and is hung in a 32-degree room which is not allowed to rise above 38 degrees. This room temperature is reduced to 32 degrees in 12 hours.

Refrigeration. The calculation of the refrigeration load is usually by rule of thumb. The daily kill [16] is a favorite method, as one ton of refrigeration is required in the ratio of the following number of animals slaughtered daily:

5 to 6 cattle at 700 lb.;
40 to 50 calves at 80 lb.;
15 to 22 hogs at 225 lb.;
50 to 65 sheep at 60 lb.;
1000 to 1300 chickens at 3 lb.

This rule takes for granted that the rooms are properly insulated. Where the piping is 2-in. direct expansion the packing house rule is:

Warm beef cooler	requires 1 ft.	of 2-in. pipe	per	10	cu. ft.	of storage.
Beef cooler	"	"	"	12	"	"
Sheep cooler	"	"	"	12	"	"
Hog cooler	"	"	"	8	"	"
Sausage (at 30 deg. F.)	"	"	"	10	"	"
Curing cellars (36 deg. F.)	"	"	"	18	"	"
Freezers (15 deg. F.)	"	"	"	6	"	"
Freezers (0 deg. F.)	"	"	"	2½	"	"

[16] Accepted practice, authority unknown.

A rough rule requires one ton of refrigeration per 7000 to 12,000 cu. ft. of space at 33 to 38 deg. F. As an illustration of the working of these rule-of-thumb rules, the following may be taken as an example:

Find the refrigeration required for 100 cattle daily killed, and hanging space for 400 cattle, 6500 sq. ft. of cooler space required with 33 ft. ceiling or 143,000 cu. ft. of space; for 250 hogs daily or 750 hanging space allowing 3000 sq. ft. at 18 ft. ceiling or 54,000 cu. ft. and a storage for a total turnover in 60 days (storage for 1,500,000 lb. of meat product)

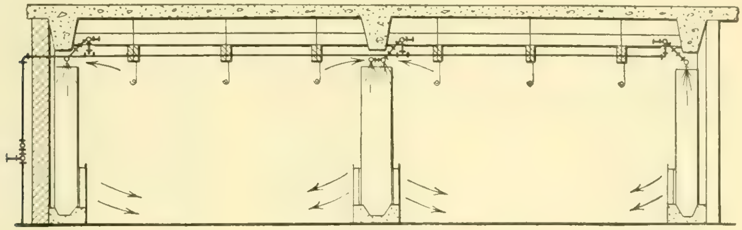


FIG. 294.—Air Cooling with Brine Sprays.

requiring 15,600 to 20,000 sq. ft. of floor and 200,000 cu. ft. of space. As a by-product there will be 10,000 lb. lard daily.

	Tons
143,000 cu. ft. of beef cooler at 1 ton per 10,000 cu. ft.	14.3
100 cattle daily at 500 lb.	12.2
54,000 cu. ft. hog cooler space at 1 ton per 10,000 cu. ft.	5.4
250 hogs at 180 lb.	10.9
200,000 cu. ft. cellar space at 1 ton per 12,000 cu. ft.	16.6
15 tons of ice at 2.0 tons of refrigeration	30.0
10,000 lb. of lard (4 hours' duration)	12.0
100,000 cu. ft. freezer space at 1 ton per 3000 cu ft.	35.0
Total	136.4

In all likelihood rough rules like the preceding have been satisfactory inasmuch as the refrigeration supplied would be much greater than the actual requirements in order to permit the installation of sufficient spares to insure continuity of operation in the time of breakdown. It seems likely, however, that in the future the refrigeration requirements will be more carefully calculated, i.e., a determination of the heat leakage and the live load.

Brine sprays have been used in the packing plant for years with varying success. The older designs used a semi-solid spray which provided circulation but was not efficient. During the chilling process a large amount of steam is thrown off into the air, the shrinkage of weight

being usually $1\frac{1}{2}$ to 2 per cent by weight of the carcass, but sometimes as much as 3 or 4 per cent. The packer desires to reduce this loss to a minimum because of the loss of weight of the product, as well as the difficulty met with at times from the large amount of frost accumulation on the pipes. The chill room is always made with a loft in order to get a lively circulation of air, and circulation is always required if quick cooling is in demand. Such lofts are shown in Figs. 292 and 299, giving an example of the arrangement of piping and sprays.

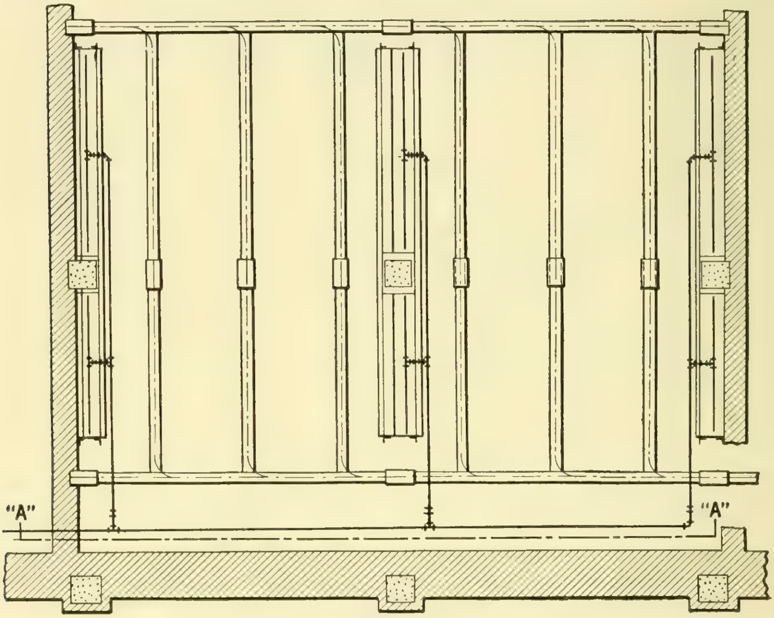


Fig. 295.—Air Cooling with Brine Sprays.

The Spray System.—The best results with sprays[16a] are obtained with atomizing nozzles. With such sprays, hogs dressed at 225 lb. can be chilled to inside ham temperatures of 36 deg. F. in 24 hours with 25 deg. brine, using 30 lb. suction pressure at the compressor. Beef can be chilled to from 34 to 36 deg. F. (inside) in 36 hr. using weak brine or possibly *water sprays* at 33 degrees and at a great saving in the shrinkage. The great advantage in the use of sprays is the quick removal of the moisture in the air, the rapid circulation and the ready capacity variation by decreasing the nozzle pressure, or the brine or the water temperature.

[16a] Samuel Bloom, The National Provisioner, February, 1924. American Society of Refrigerating Engineers, Journal, 1921.

The objection is the shrinkage and the danger of the mists carrying over and being deposited on the meat. Up to the present, water in the sprays of chill rooms has not been found practical, and the result is that *salt* brine (NaCl) is always used. In order to decrease the shrinkage the humidity is kept as high as possible, requiring a low salt concentration and therefore a relatively high temperature of the brine.

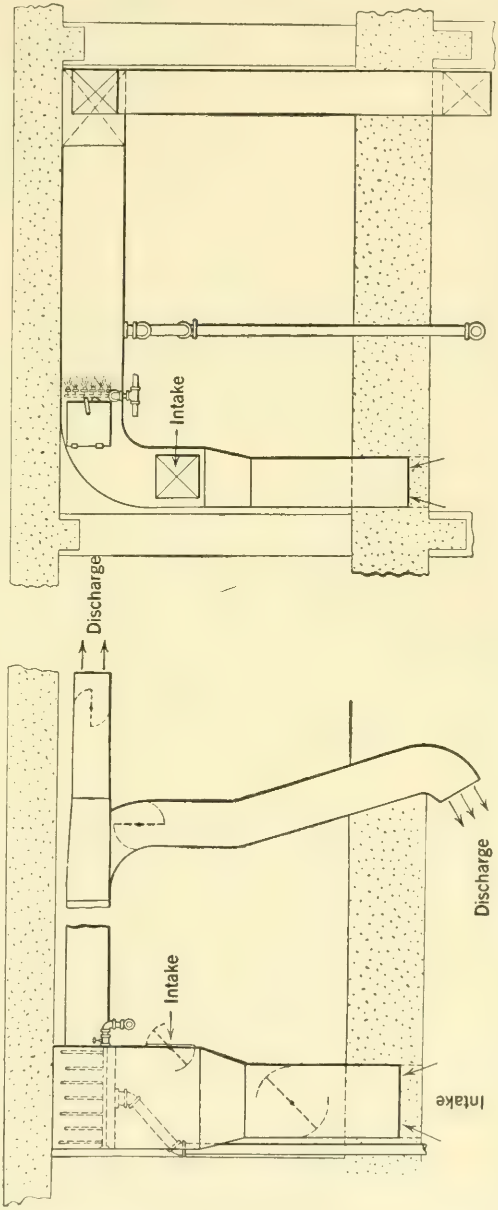


FIG. 296.—Air Cooling with Brine Sprays.

Mr. Bloom advocates that the height of the loft be made proportional to the load, the width of the deck and the temperature of the brine, but he considers that it should never be over 6 ft. nor less than 2 ft. (because of insulation difficulties and waterproofing of the deck). The width of the two openings in the deck should be equal to the loft height, and the deck should be insulated with 2 in. of corkboard which should be waterproofed very carefully.

The difficulty with dirt in the brine is overcome by the operation of leaching the salt on solution, and by having settling tanks which may be cleaned out every year. The

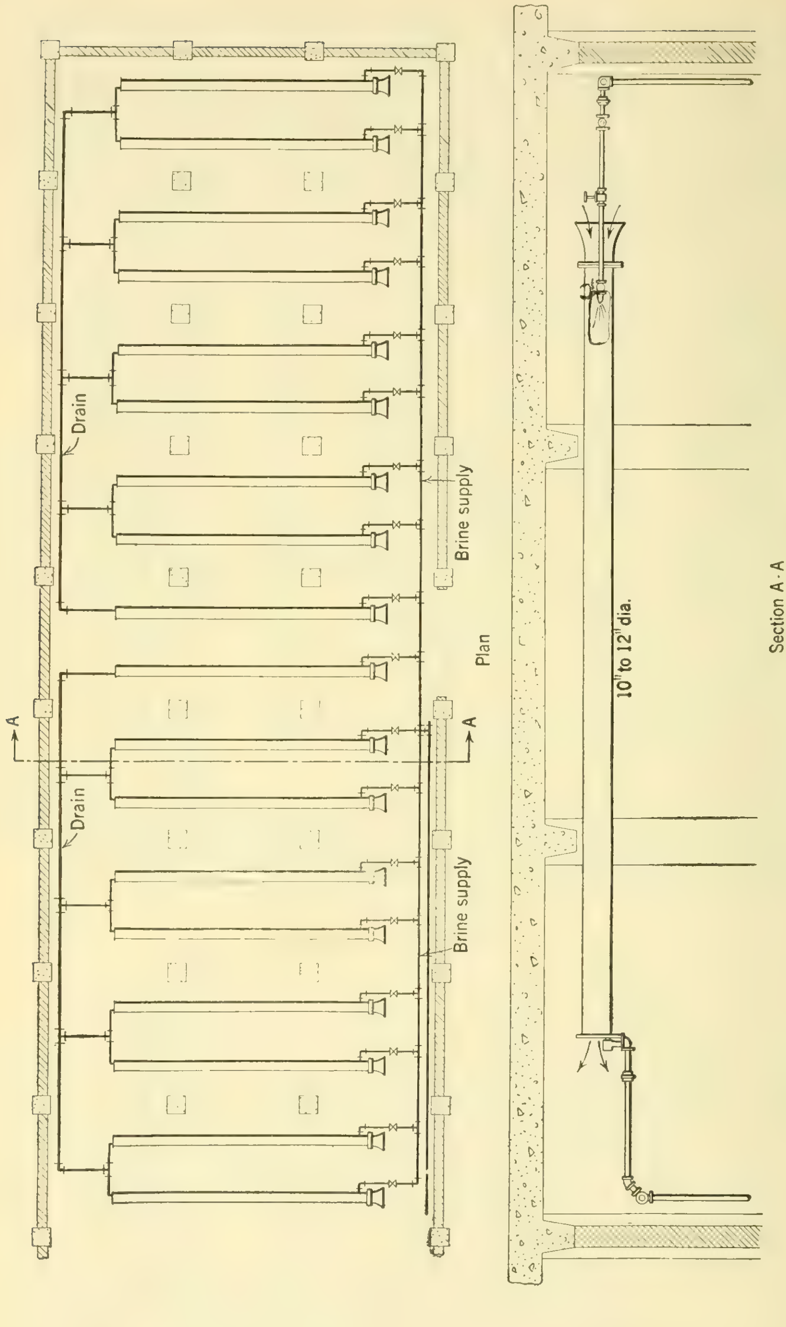


FIG. 297.—Air Cooling with Brine Sprays.

brine pressure at the nozzle should be from 9 to 2 lb. per sq. in. minimum and the decks should be from 38 ft. to 9 ft. respectively. Too narrow a deck will permit the droplets of brine to carry over, for even if they are in the air only 3 seconds at a velocity of 600 ft. per min. the distance carried will be 30 ft. The height of the spray above the deck and the velocity of the air are the determining factors as regards the width of the deck.

To cool the brine the shell and tube brine cooler is the preferred means because of its great flexibility, high heat transfer and relative lack of resistance to brine flow if the number of passes are small. The shell and tube as well as the double pipe brine cooler must be watched carefully to prevent injury due to freezing of the brine should the pump stop. These coolers when placed on the *suction* side of the pump can be made safe provided a means of draining the cooler is arranged for in the event of the stopping of the pump. Calcium chloride brine is never used for sprays in the packing house because of the drying action of the calcium brine, because of the greater cost and of the danger of the mists carrying over and being deposited on the meat, thus destroying palatability. The best arrangement is to provide a float-operated tank in order to keep a constant head on the nozzles. If a centrifugal pump is used it should be such a one as will give a " flat " characteristic curve—enabling one to get the same pressure on the nozzle for a range of capacity of the pump (see Chapter VIII).

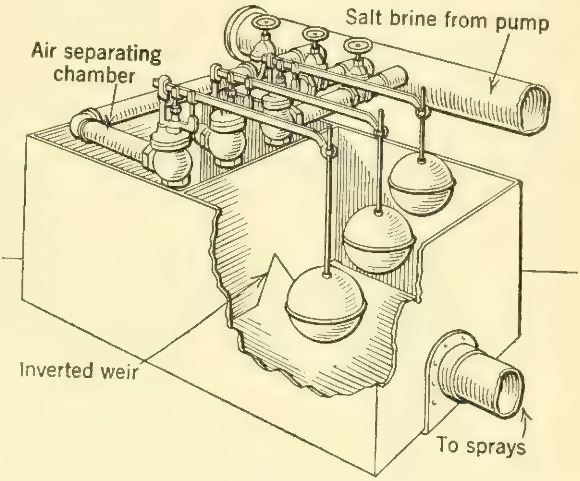
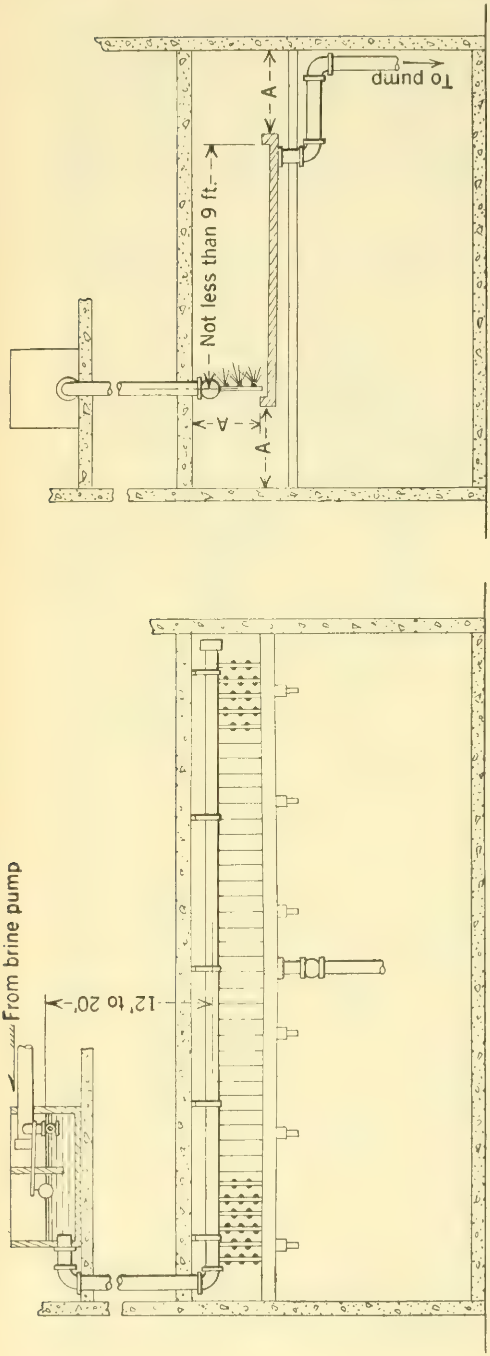


Fig. 298.—Gravity Brine Tank.

Problem.—As an example when an attempt at a close calculation is to be made, let it be required to chill 50 beeves at 750 lb. each from a temperature of 90 deg. to 38 deg. F. in 12 hours. The brine in the fore-cooler will enter the spray at 32 degrees



Press. at Nozzle lb/sq. in.	Gallons per Minute		
	Webster $\frac{3}{16}''$ open	Braemer #30	Carrier $\frac{3}{16}''$ open
6	1.00		0.88
9	1.25		1.10
12	1.41	1.55	1.33
21	1.87	2.05	1.84
24	2.00	2.19	1.98
27	2.12	2.32	2.09
30	2.24	2.45	2.20
33	2.34	2.57	2.32
36	2.45	2.60	2.40
39	2.55	2.80	2.50

FIG. 299.—The Gravity Brine Tank.

and will rise in temperature to 35 degrees. The regular spray deck will be used, and salt brine only at as low a concentration as possible will be made use of.

To cool the meat	$50 \times 750 \times 0.77 \times (90 - 38)$	=	1,501,500 B.t.u.
		=	10.43 tons
Allowing 20 per cent for losses (these could be calculated if the details of the insulation, etc., were known		=	2.09 tons
	Total	=	12.52 tons

Allowing 12 sq. ft. per animal there will be required $12 \times 50 = 600$ sq. ft. floor and a room 20 ft. by 30 ft. will answer the conditions. The usual distance between rails is 4 ft. 0 in., and the distance from the rail to the floor 11 ft. 2 in. (Fig. 299). The amount of brine required will be as follows:

$$\text{Wt.} \times 0.892 \times 3 = 12.52 \times 200 \qquad = 2504 \text{ B.t.u. (per minute)}$$
$$\text{Wt. of brine per minute} = 2504 \div 2.676 = 936 \text{ lb.}$$

Using a specific gravity of the brine of 1.066, the gallons of brine required will be $936 \div (8.33 \times 1.066) = 105.4$. Under a head of 13.05 ft. (6.0 lb.) the No. 30 Webster nozzle will deliver 1 gal. per min. and there will therefore be required 105 nozzles. The maximum spacing of the nozzles is 20 in., so the placing of these four high on 10-in. centers, and 26 stands equally spaced horizontally (less than 14 in. apart), will be a good distribution. The height of the deck will be 3 ft. 6 in., and this will also be the size of the two openings in the deck. This will provide a vigorous circulation of the air of about two changes per minute, there being no choking at any point.

Lard and Compound Cooling.—Lard weighs about 59.3 lb. per cu. ft., the specific heat is from 0.5 to 0.6 depending on the fluidity, and the latent heat is 90 B.t.u. per lb. The melting point is usually understood to be 70 deg. F. for lard and 105 deg. for tallow.

Lard compound, a mixture of hog fat with beef fat and cottonseed oil, has a specific heat of from 0.3 to 0.5 depending on its fluidity, a latent heat of 90 B.t.u. per lb. and a specific gravity of 0.92 (57.5 lb. per cu. ft.).

Mechanical refrigeration is used in the cooling of both lard compound and lard. The manner of applying refrigeration is to use hollow cylindrical drums, which are maintained at a temperature of about 10 deg. F. by means of brine or direct expansion, and which are made to revolve at from 10 to 14 r.p.m. The usual custom is for the lard or the compound to be cooled first with water from 135 to 140 deg. F. to 80 degrees and then by passing the commodity to the cooling rolls to cool it to the final temperature of from 30 to 40 degrees.

In the manufacture of lard the object of refrigeration (besides the speed of the operation which is very important) is the uniformity and better quality of the product. In the manufacture of lard the trimmings

and other fat parts of the hog are steamed in a rendering tank under 40 lb. steam pressure for some 7 to 9 hours. It is then mixed with fuller's earth for a few minutes, and next filtered by passing it through a filter press. The resulting hot, white lard is stiffened by adding not more than 5 per cent of lard stearine, and then by cooling mechanically.

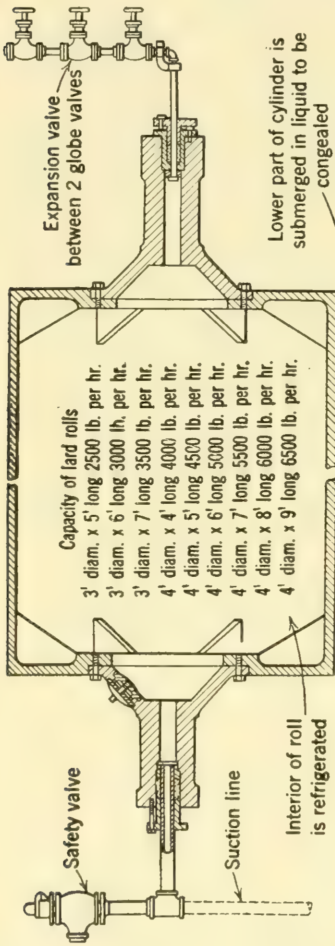


Fig. 300.—Section of a Lard Roll.

Compounds are made using beef fat, cottonseed oil and pork fat. In making such a compound the refined fats are mixed and filtered, and then sent to the cold revolving drum. The drum chills the warm fluid quickly, thereby giving a uniform compound. Should slow cooling be permitted the beef crystals would freeze out first and non-uniformity would result. The revolving drum, cooled with brine or ammonia, freezes a thin layer of lard or compound, as the case may be, and, after the commodity has been on the surface for about three-quarters of a revolution of the drum, a carefully adjusted knife scrapes it off the surface. The semi-solid substance drops into the picker trough where thorough mixing is again brought about (Fig. 301). The product is now ready for the packages. Figure 300 shows the lard cooling cylinder in section, and Fig. 302 shows the entire process. As an example of the refrigeration necessary let it be required to find the amount of cooling effect necessary to handle 10,000 lb. of lard in four hours with brine. Also let it be required to find the size of the compressor and the size of the required cooling cylinder.

Problem.

$$10{,}000 \times 0.6 \times (80 - 30) + 10{,}000 \times 90 = 1{,}200{,}000 \text{ B.t.u. per 4 hours,}$$
$$= 300{,}000 \text{ B.t.u. per hour.}$$
$$300{,}000 \div 12{,}000 = 25 \text{ tons of refrigeration.}$$

Adding 10 per cent for losses = 27.5 tons of refrigeration required. As the compressor will be required to cool brine to + 10 deg. F., a zero degree boiling temperature of the ammonia will be required, and this corresponds practically to 15 lb. gage pressure suction to the compressor. With such a suction pressure there would be required 4.63 cu. ft. piston displacement per ton of refrigeration per minute using a volumetric efficiency of 0.83 and the piston displacement for 27.5 tons

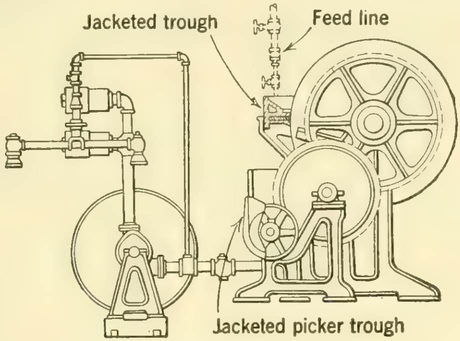


FIG. 301.—Assembly of Lard Rolls.

would be 128 cu. ft. per min. Referring to Fig. 300, which gives the construction of cylindrical drums, it will be seen that the size 3 ft. diam. and 5 ft. long will handle 2500 lb. per hour. This will be the required size. The thickness of the lard on the surface of the roll under these circumstances is less than 1/50 in.

The Refrigerator Car.—It has been found in cold storage practice that *only* the best construction is the cheapest, and that a rough rule of

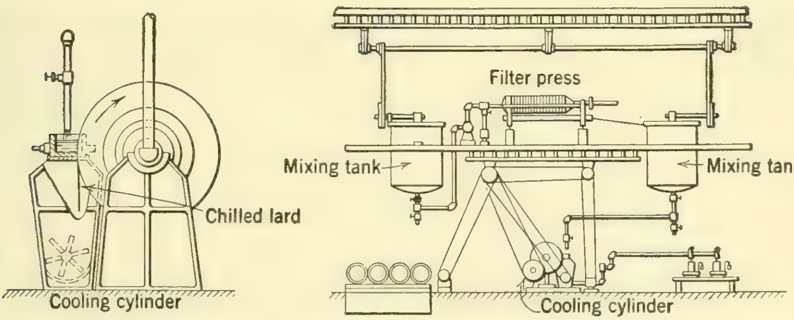


FIG. 302.—Lard Manufacture.

1 in. of corkboard or its equivalent is required for every 10 degrees of depressed temperature, and the same rule (slightly modified) applies also to the refrigerator car. For an average of 70 degrees outside and 40 degrees inside there would be required, according to this rule, 3 in. of corkboard. W. H. Winterrowd [17] states that calculated values for heat

[17] Some Notes on Railway Refrigerator Cars, Mechanical Engineering, July, 1922.

transfer for typical refrigerator constructions vary from 1.7 to 2.33 on the roof, 2.17 to 2.89 on the wall, and 2.46 to 2.54 on the floor, B.t.u. per sq. ft. per 24 hours. The value for 3-in. corkboard alone is about 2.1 without allowing for air spaces and the building construction. The

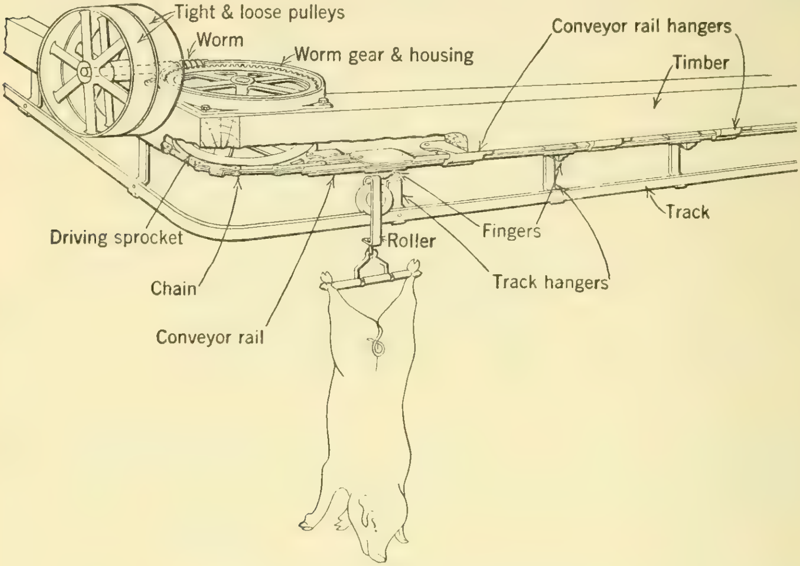


Fig. 303.—Overhead Conveyor.

refrigerator car and cold storage buildings are not comparable, however, as the car has a very rough service and the shock and vibration is such as to make it difficult to keep the car tight. The design of the car then must be one that will resist heat transfer, and will have a long life of service.

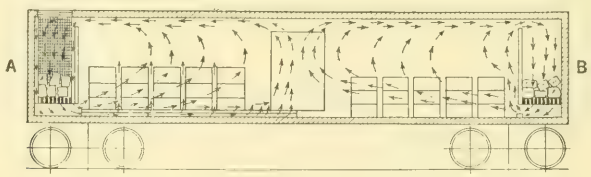


Fig. 304.—Air Circulation in a Refrigerator Car.

Next to heat transfer is the method of cooling and the efficiency of the air circulation (Fig. 304). The lading is continually developing heat which must be absorbed or cause a rise in the temperature. Cooling is applied in two principal ways: the best known and the most usual

method is the ice bunker at the two ends of the car, and the later design embodies a brine tank built into the roof at each end of the car, the two tanks being connected by pipes hung some two or three inches beneath the ceiling and so arranged that check valves in the tanks which are operated by the swaying of the car causes a positive flow of the brine. Ice and salt, of course, are charged into the tanks. The advantages claimed are more economy of space, and more even car temperatures. It is not clear what efficiency is obtained during the periods when the car is still.

TABLE 100

[By the U. S. Bureau of Agriculture Economics]

CARLOAD SHIPMENTS OF DOMESTIC FRUITS AND VEGETABLES

Commodities	Number of Carloads						
	1917	1918	1919	1920	1921	1922	1923
Apples	57,048	68,840	81,552	102,962	96,498	101,780	127,505
Asparagus	1,057	1,120	1,101	1,034	...	746	
Cabbage	20,354	28,661	24,982	31,020	31,718	40,065	36,395
Cantaloups	17,430	13,619	22,039	22,377	25,569	29,917	25,839
Carrots	...	...	...	...	...	1,987	
Cauliflower	...	1,535	2,546	3,410	12,483	14,151	16,624
Celery	6,577	7,412	6,449	9,308	3,895	3,991	4,714
Cherries	897	680	1,025	1,870	...	2,447	
Cranberries	...	...	...	...	...	2,009	
Cucumbers	3,711	2,757	4,306	...	...	6,151	
Eggplant	...	300	136	...	...	271	
Grapes	21,379	20,915	30,349	39,205	37,202	59,858	64,020
Lettuce	5,428	6,959	8,018	13,818	18,616	22,840	29,493
Onions	19,152	22,027	20,874	25,950	23,319	27,563	26,747
Peaches	27,237	20,409	30,923	26,967	27,300	38,291	33,630
Pears	11,468	10,311	10,158	14,950	12,821	20,138	18,144
Green peas	...	691	278*				
Peppers	...	1,170	913	...	...	2,322	
Plums and prunes	...	3,125	3,222	4,430	...	5,021	6,786
Sweet potatoes	8,898	9,254	13,725	16,254	19,071	20,576	19,793
White potatoes	144,656	169,462	181,277	179,149	219,426	245,221	238,983
Spinach	1,278	2,422	2,957	2,687	4,924	5,205	7,682
Strawberries	15,065	8,452	8,105	8,490	10,695	18,716	17,775
Tomatoes	14,115	15,471	14,503	15,556	17,199	26,668	24,002
Turnips	...	...	...	...	...	835	
Mixed vegetables	...	8,804	11,933	12,740	15,816	19,676	24,016
Watermelons	31,503	20,392	30,860	39,255	46,463	47,066	35,405
Deciduous fruits:							
Mixed	...	...	...	...	...	8,609	8,998
Citrus fruits:							
Grapefruit	393	5,650	6,624	12,086	12,275	14,182	19,150
Lemons	401	6,913	8,823	9,718	11,887	9,875	8,668
Oranges	5,437	28,444	49,324	53,041	65,891	46,526	74,948
Miscellaneous	...	3,351	3,024	3,902	5,716	2,088	
	413,484	489,236	580,026	650,179	718,784	844,191	866,317

* Incomplete.

The following paper on "Railway Car Icing Systems" was presented by A. L. Blatti before the annual meeting of the National Association of Practical Refrigerating Engineers, held in Memphis, December 12 to 15, 1923.

It has been found that a platform longer than 1000 ft. is not practicable in some cases, because the time consumed in moving ice on such a platform more than offsets the advantage of having more cars at the dock than a 1000-ft. platform will accommodate. A platform of 1000 ft. length will accommodate 46 cars, or 23 cars on each side, and when more cars than this number arrive in a single train, it is customary, after splitting the train into two groups of cars which are placed on each side of the dock, to ice the group having the surplus cars first, and then while icing the other group of cars on the opposite side of the dock, the switching crew moves into place the hang-over or surplus cars, which are in turn iced without cessation of the entire operation or delay in the work of the icing crew.

Our experience has been that if the cakes of ice employed are of 300-lb. size, the icing platform should be 14 ft. wide, while if a 400-lb. cake is used it should be 16 or 17 ft. wide. The platform should be 14 ft. 6 in. above the top of the car rail, so that there will be sufficient fall from the platform to reach the far side of the bunker opening by means of a skid which will clear the running board on top of the car. The platform should, of course, have a conveyor chain running its entire length through its center and operated by reversible motor, so that ice may be moved backward and forward at will to any selected place on the platform.

The platform described is for use in placing ice in the bunkers of cars which contain fruit or vegetables and, therefore, do not require temperatures approximating the freezing point, so that cake ice may be employed. The icing of cars which contain packing-house products demanding a lower temperature, requires a different process, as crushed ice mixed with salt must be used for this purpose, the quantity of salt ranging from 10 to 15 per cent by weight of the crushed ice. For meat car icing a second deck to the platform previously described may be constructed, 7 ft. above the top of such platform. This upper deck will have no conveyor chain, but will provide a run-way for carts holding from 900 to 1000 lb. of crushed ice each, which carts may be moved upon the upper deck to the position required and the crushed ice contained in those carts dumped into movable traveling chutes, which are attached to the edge of the upper deck and which chutes are moved into place so that the lower end of the same point directly into the bunkers of meat cars being iced. Salt bins should be provided at intervals on the lower deck, so that they are readily available to the icing crew who may shovel such salt as is required into the bunker of the meat cars, after the crushed ice has been dumped into the bunker through the chute from the upper deck.

The average time required to ice vegetable or fruit train with cake ice varies from 1 to $2\frac{1}{2}$ minutes for each car, according to the amount of ice required in each bunker, while for icing of meat cars with crushed ice $2\frac{1}{2}$ minutes per car is a good average time because of the fact that additional time is required in draining the ice compartment of a meat car and working down the ice which may remain in the same, preparatory to the reception of additional crushed ice.

Considerable attention has been given to details in car construction during the last ten years and particularly since 1917. Figure 304 shows in "*A*" the basket type and in "*B*" the regular bunker type of box

bunker. The latter is now obsolete on account of the obstruction to the free circulation of air, poor cooling of the air and the resulting higher air temperatures in the car. The *basket* type is made of wire mesh with a slatted wooden structure, so arranged as to permit air to flow all around the basket. Referring to Fig. 304, it will be seen that the left-hand side has a false floor. This rack is arranged so as to give some 3 in. to 4 in. of space for the flow of air and is arranged with hinges at the sides of the car and is divided along the middle of the car so that it can be swung up against the wall. The probable air circulation is shown

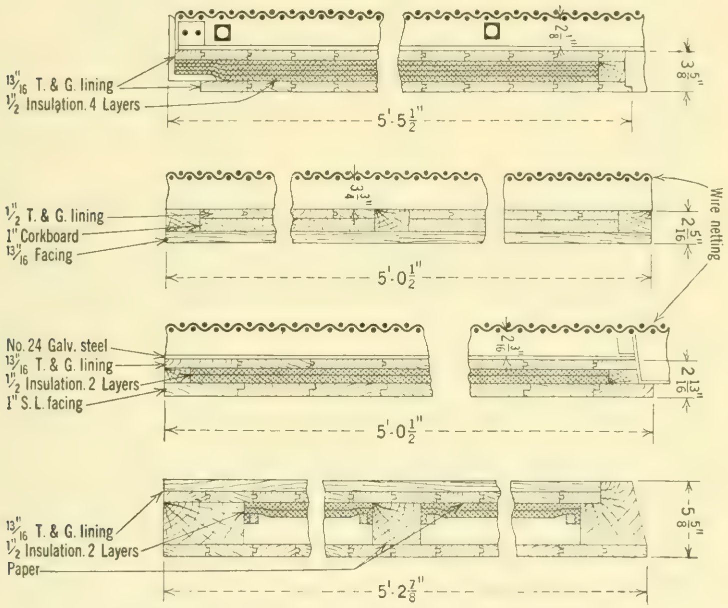


FIG. 305.—Insulation of Refrigerator Cars.

for the types, the slatted design giving the better results as would be expected since air can travel to the middle of the car whereas it becomes short-circuited with the construction shown on the "*B*" side of the car. The center of the car is the most severe test of refrigeration, and the length of the car is an important factor in the refrigeration efficiency. The general tendency is to have a length between bulkheads of from 32 to 34 ft., some 6 to 10 in. from the rack to the bottom of the opening in the bulkhead, and from 12 to 18 in. from the top of the bulkhead to the ceiling. The construction of the bulkhead is important, for otherwise freezing of the lading might occur near it. Figure 305 gives an idea of the modern trend.

The construction of the walls, roof and floor is given by Figs. 304 to 307. In the paper by W. H. Winterrowd it was brought out that corkboard is used in but few cases, except on the floor where 2-in. corkboard is usual. The feeling seems to prevail that corkboard does not endure

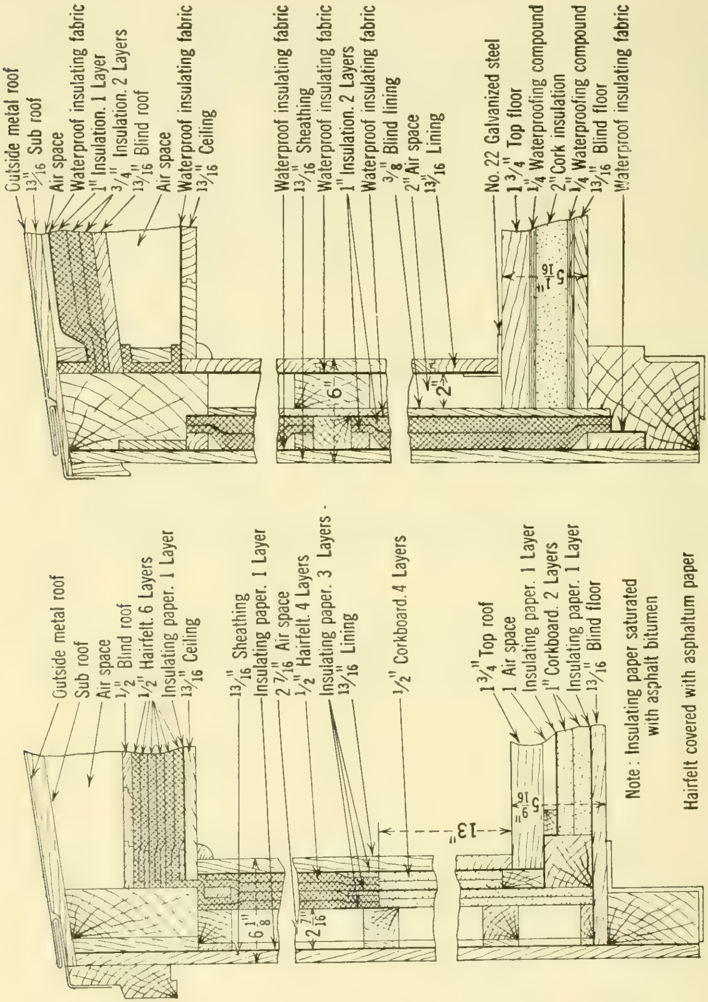


FIG. 306 and 307.—Insulation of Refrigerator Cars.

as well as more flexible insulation, but that the insulation must be suitably protected from moisture. The floors and the walls should be constructed so as to permit the driving of nails without injury to the insulation.

REFRIGERATOR CARS IN OPERATION

Information Bulletin, No. 345, of the Perishable Freight Conservation Bureau, American Association of Ice and Refrigeration, quotes some figures from railway equipment register concerning the number of refrigerator cars in operation. The figures are as follows:

U. S. Railroads	32,773
Car Lines (Railroad controlled)	94,007
Car Lines (Independent)	5,855
Meat Packers	18,650
Canadian Railroads	6,093
Mexican	2
Total	157,380

Installed in Service:

	1923	First 9 mos. 1924
Railroads	7,570	3,801
Private	14,508	7,875
Total	22,078	11,678

According to the figures for the first nine months of 1924, there will be quite an appreciable decrease in the number of cars installed in service for the year 1924.

Mechanical Refrigeration.—Many attempts have been made to refrigerate freight cars mechanically. However, the difficulties are so many that it seems improbable that success will be obtained for American cars for some time. A mechanical drive from the axle seems to be impossible, considering the frequent delays. The central plant car to be designed for ten or more cars could be operated by a standard oil engine, the engine and compressor to be air cooled, probably using a water cooled condenser and an automobile type of air cooled radiator with improved fan ventilation for cooling the water. This would enable the water to be conserved and at the same time would permit better heat transfer in the condenser..

Precooling of Fruits and Vegetables.—Most fruits and some vegetables are shipped by means of refrigerator cars from three to six days' carry. If the fruit is warm (say 80 to 90 deg. F.) when loading into the freight car it takes from 3 to 5 days[18] or more to get the temperature of the inner packages down to a temperature of 40 to 45 deg. F. and in many cases this degree of temperature is never reached, while in any case the temperature is very uneven throughout the car. Much fruit is

[18] Bulletin No. 123, Bureau of Plant Industry, Department of Agriculture.

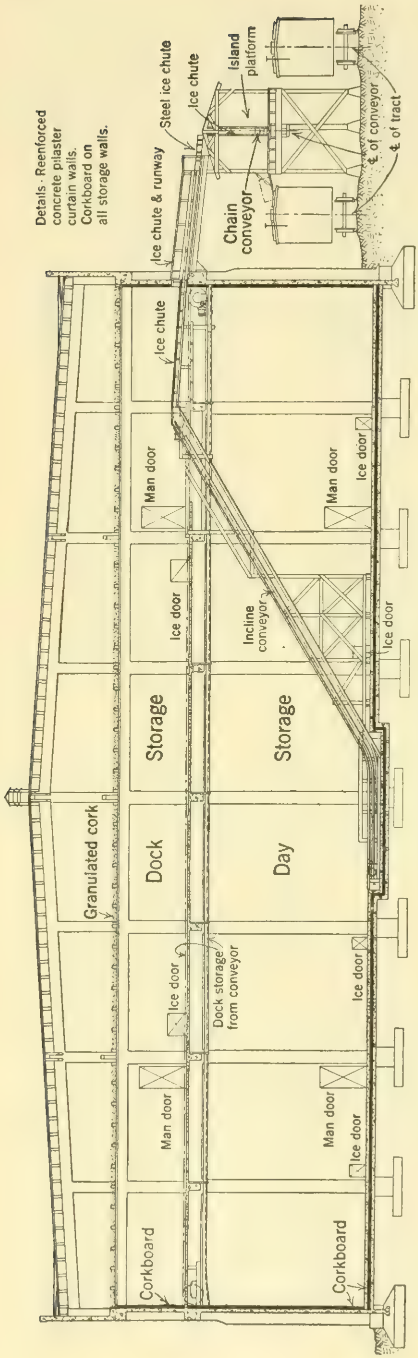


Fig. 308.—Refrigerator Car Icing.

spoiled, or the upper layers of packages have a secondary quality as compared with those on the floor near the ends of the car. Not only does the fruit spoil by the action of the fungi, but the resulting formation of carbon dioxide tends to increase the temperature. For example, it is stated that peaches at 86 deg. F. will rise 18 degrees in 20 hours, while at 68 degrees the increase is only 6 degrees in the same interval of time. A rough calculation gives an ice meltage of 22 per cent due to the self-heating of the fruit. It is evident then that shipment of perishable fruits involves the following important phases:

a. The development of a high grade refrigerator car.

b. Means of properly precooling perishable fruits and vegetables.

c. Economical means of car icing at convenient points of train departure and suitable re-icing depots at division points along the road.

The importance of proper precooling or icing or both can be well brought out by the history of the expansion and development of the Pacific Fruit Express Company as an example of what the industry has done in that respect in the last 20 years.

In 1907 the Pacific Fruit Express Company (operated jointly by the Union Pacific and the Southern Pacific Railroads) owned 6600 refrigerating cars, handling 48,900 carloads of perishables of which 50 per cent were refrigerated. In 1921 these had increased to 19,200 cars, which handled 170,000 carloads of perishables, of which 70 per cent were iced and which required for the purpose 1,600,000 tons of ice. At first a large amount of the ice used in the car bunkers was natural ice, but this was found to be uneconomical and not dependable, so that in 1921 about two-thirds of the ice was manufactured. In 1922 the Pacific Fruit Express Company had a daily capacity of 3500 tons of ice making and storage capacity for 230,000 tons located at 12 ice plants (that is in addition to 5 natural ice plants having 100,000 tons storage capacity),

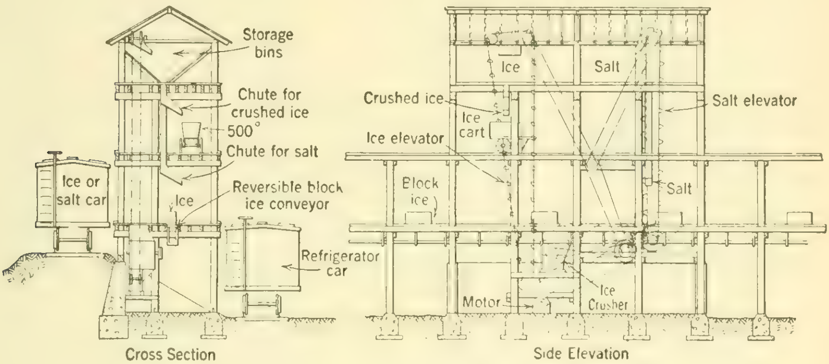


FIG. 309.—Refrigerator Car Icing.

and had 1300 tons daily ice making and 80,000 tons storage capacity under contract with private concerns. The separate plants vary in size, that at Roseville, California, having a daily capacity of 800 tons and a storage capacity of 46,000 tons, and during 1921 using a total of 165,000 tons of ice at this point with a maximum for one day of 2400 tons. The standard icing platform is the island type (Fig. 308) for 55-car trains on each side. The platforms, which are of wooden construction, are 14 ft. high and 14 ft. wide, and the ice is moved by means of an endless chain conveyor. The train is iced during the time of inspection and change of engine and caboose. The procedure in California (1924) in loading a refrigerator car is as follows: the car is cleaned thoroughly and receives its load of ice and salt. It is then sent to the point of shipment, receives its load of perishables, and is brought back with all possible speed to the car icing division point and is made up into trains where enough ice is added to carry it to the next icing point. From

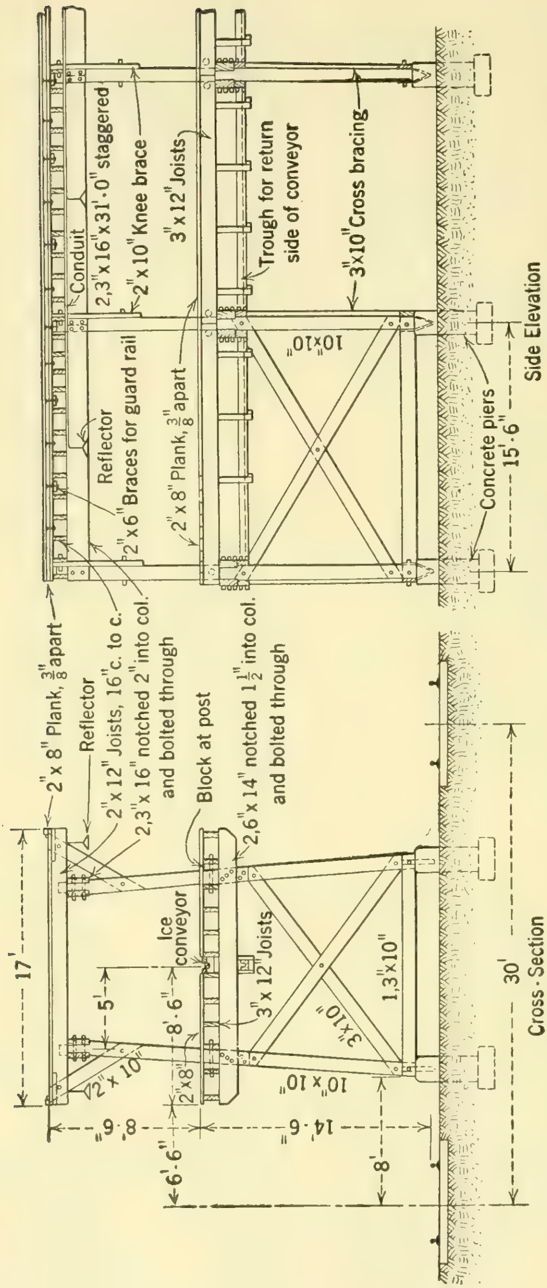


FIG. 310.—Refrigerator Car Icing.

5 to 6 tons of ice are required to ice a freight car. Figures 308 to 311 show ice and salt loading platforms and devices.

Precooling.—The development of the precooling industry has been rapid because of the failure in accomplishing a quick chilling of the lading in the ordinary refrigerating car. At the present time (1927) there are three principal means of precooling. First, there is the shippers' precooling design where the lading is placed in cold storage rooms in which the proper temperature is carried and the temperature of the lading is dropped as quickly as possible by the use of fan circulation of the air or otherwise. After the proper temperature has been reached, the lading is loaded into precooled refrigerator cars, having been iced a sufficient length of time previously to cool the car properly. The second means of precooling is the carriers' precooling design. By this is meant that the car is loaded with warm goods after which the entire car and lading are cooled mechanically. Such means can be used only where large amounts of perishable produce (such as citrus fruits) originates, as for example, near San Bernardino, Calif., and Zellah, Wash. In these plants the cars are placed in a precooling dock, the ventilator hatches are connected with large air pipes and the lading is cooled by means of forced circulation of cold air. Finally there is the third method of precooling certain vegetables, including lettuce. Here, water at 35 deg. F.

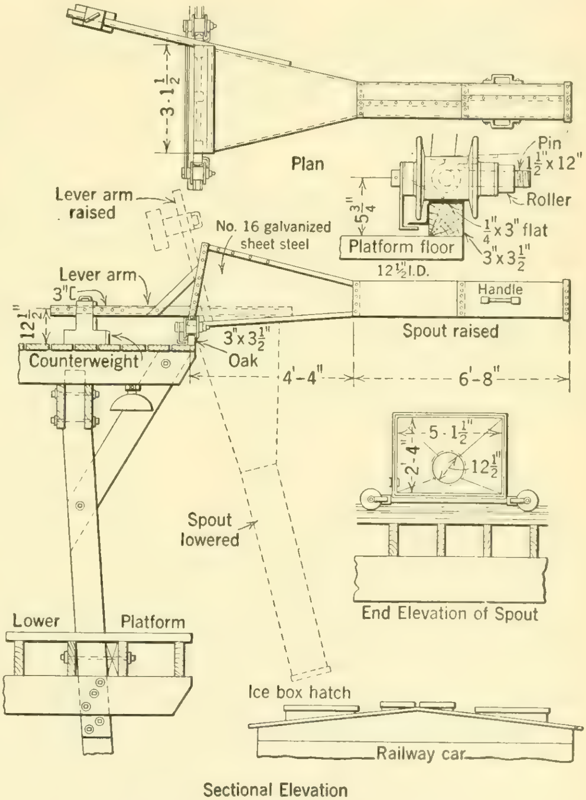


FIG. 311.—Refrigerator Car Icing.

is arranged to flow over the produce and cooling is accomplished very quickly. In the paper by J. W. Andrews[19] the water is cooled by means of Baudalot coolers. The crates of celery are made to pass through a pan of water at a temperature of from 33 to 35 deg. F., and in 20 min. some 70 gal. of water have passed through the crate, after which the surplus water is removed and the celery is delivered to the refrigerator car door at or near 35 degrees temperature. A carload of celery can be cooled and stored in the car in $1\frac{1}{3}$ hours. This means of cooling by direct contact appears to be efficient and economical and seems to improve the product.

Shippers' Precooling.—In the state of Florida there were in 1924 15 plants with an annual shipment of 4000 cars of perishables. In this locality salt (NaCl) brine is used for cooling the air, using a spray chamber of about 15 ft. by 9 ft. by 9 ft. high and air velocities of about 200 ft. per min. The usual eliminator removes the brine mist and the air is then passed through overhead ducts designed for 1000 ft. per minute velocity. Reversing of the air flow is provided for usually by reversing the motor drive on the fan. The air is forced into the room (Figs. 312 and 313) where the crates are closely packed, and the air travel is directed by means of baffle curtains "*B*" at the top of the room, resulting in an air pressure drop of from 0.14 to 0.25 in. of water, and there is stimulated a fairly uniform flow of air through the boxes. The inside box temperatures are also given in the figure.

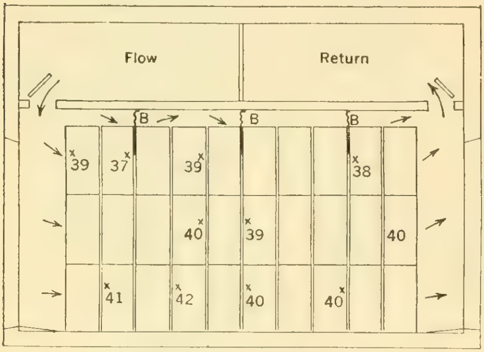


FIG. 312.—Precooling.

The results of tests according to the paper by J. W. Andrews show that the average initial temperature was from 73 to 77 deg. F. Cold air circulation was applied for $10\frac{2}{3}$ hours to get an average temperature of the crates of 40 degrees. The temperature of the air circulated was 36 deg. F.

The Carriers' Precooler.—The cooling of the car and the lading with cold air is known best as the Gay process. In his experiments Norman Gay found that all freight cars, even when closed as tightly as possible,

[19] J. W. Andrews, Precooling of Fruits and Vegetables, Refrigerating Engineering 1924.

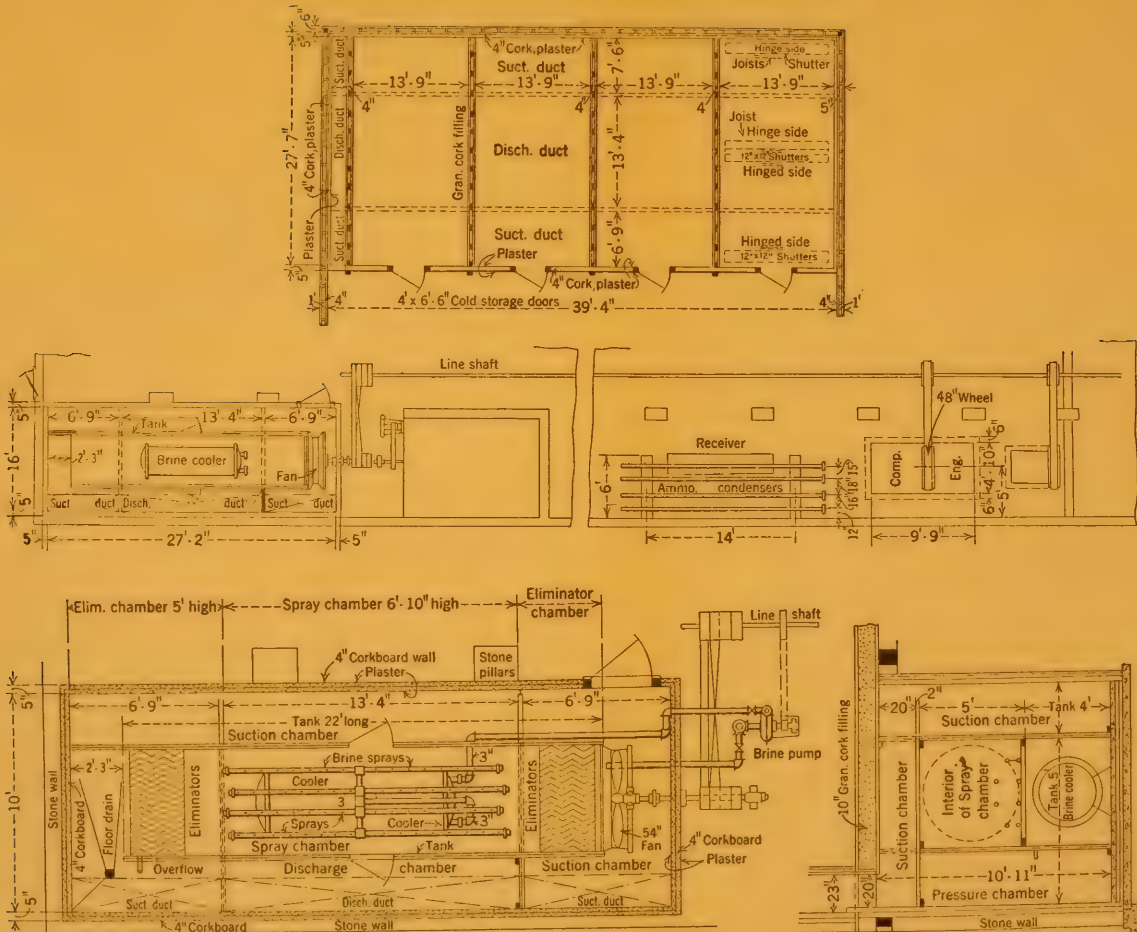


Fig. 313.—Precooling with Brine Cooled Air.

To face page 488.

would permit a loss of from 800 to 2000 cu. ft. of air per minute under $\frac{1}{2}$ in. static pressure and the same loss was obtainable with $\frac{1}{2}$ in. vacuum. Therefore any system whereby air is to be forced into the car would need to be " balanced " as far as the pressure in the center of the car is concerned. To do this a pressure fan as well as an exhaust fan is required and adjustment is made so that on opening the center door no blast is noticed in either direction although from 6000 to 8000 cu. ft. of air per minute is being forced through the car. At the San Bernardino plant the lateral flexible pipes had an internal diameter of 22 in., and 32 cars connected together at one time, requiring each 8000 cu. ft. of air per minute, will be cooled within 4 hours at the maximum and fully chilled and iced in 5 hours. The air in the plant was moved by 8 No. 11 Sirocco fans of a total of 300,000 cu. ft. per minute capacity, and this air was cooled in a brine bunker room. The brine itself was cooled by double-pipe brine cooler of 200 tons capacity. The San Bernardino plant stopped the use of the Gay system in 1924.

Marine Refrigeration.—Marine refrigeration is an application of the art of cold storage peculiar to itself. It is, of course, restricted by the limitations imposed on all marine engineering design and, in addition, the ship is usually subjected to the condition of one-way traffic—at least in the respect of refrigerated cargoes. The result is that refrigerating vessels need to be designed so as to take general cargoes, and the hatchways must be made large enough for the purpose. In general, a well designed banana carrier (as far as refrigeration is concerned) will be satisfactory also for carrying citrus fruits, but the longer the trips the faster the ship's speed for economical results. However, a ship designed for carrying mixed frozen and chilled meats would not be satisfactory for frozen meats entirely, nor for a general cargo, inasmuch as fruits, for instance, require ventilation.

Insulation.—Insulation[20] usually is corkboard, granulated cork, or sometimes mineral wool which may be mixed with granulated cork. When the insulation is all cork, the sides and the overhead consist of an exposed finished layer of wood boards, two layers of 2-in. corkboard, one layer of boards and paper with the space between them, and the ship's side filled with granulated cork.

When mineral wool is used as the main insulation it consists of wool between an exposed finished layer of wood boards and the ship's sides of the deck. For the intermediate and the weather decks the insulation is placed underneath, whilst for the tank top or the floor of the hold it has to be capable of carrying the weight of the cargo. In these latter

[20] A. H. Baer, Journal, American Society of Refrigerating Engineers, 1919.

cases the insulation is usually from 5 to 6 in. of corkboard laid in pitch and with a finishing layer of boards.

Marine refrigeration[21] has four kinds of refrigerating equipment.

1. *Fruit Carriers* need forced indirect air circulation (also external fresh air ventilation). Bananas, for example, are loaded in bulk but are not dunnaged although they are divided by means of bins.

2. Frozen meats require no ventilation but the space is heavily insulated. The cargo is loaded in bulk, suitably dunnaged for natural air ventilation. This cargo should be kept at from 12 to 15 deg. F.

3. Chilled meats require a minimum height in the storage space, which is heavily piped under the ceilings, and the meat to be supported by the deck girders by means of the usual meat hooks running on rails. As temperature changes are not permissible beyond a small limit, the piping controls must be good.

4. This sub-division includes general cargoes. Frozen meats may be stored in the lowest hold. However, this division is a combination of the other three.

Banana carriers require considerably more refrigerating capacity than meats and other cargoes because as a rule the banana is not precooled, but the cargo and the hold should be precooled whenever possible. The ship should be capable of carrying apples and citrus fruits. The temperature required for bananas is 53 to 55 deg., for oranges 42 to 45 deg., for lemons 55 deg., and for apples 32 to 33 deg. F., with particular attention to the ventilation. The cargo should be raised off the decks by means of dunnage battens and (if the fruit is in packages) every alternate row should be vented vertically so as to insure proper air circulation throughout the bulk of the cargo. Bananas are stacked vertically, uncrated, usually two rows high and a third row lying flat unless the voyage is greater than 10 days. The decks are insulated, preferably, on the under side, and care should be taken to prevent construction which interferes with this ventilation. The refrigerating load consists of the heat leakage, the cooling of the cargo and the cooling and the dehumidifying of the fresh air required for ventilation and to keep the carbon dioxide content of the air in apple storage at 10 per cent or less. Figure 314 gives a good idea of the methods used in such work. Bunker rooms appear to be used entirely, contrary to the practice in stationary work, where the water or brine spray is used frequently.

The Food Investigation Board of the British Department of Scientific and Industrial Research has recently published Special Report No. 20, entitled "The Problems of Apple Transport Overseas," which is a gen-

[21] Llewellyn Williams, Naval Arch. and Marine Eng., November, 1924.

eral survey and summary of the results obtained by a scientific expedition to Australia in 1923.

A synopsis of the report is given as follows:

The Australian apple export season of 1922, although a record in point of the number of cases shipped, was in some ways one of the most disappointing in the history of the industry. Of the total export of approximately 2,000,000 cases, it has been estimated that between 500,000 and 600,000 arrived in England with the fruit content in a severely damaged condition. From three of the largest consignments, totalling altogether nearly 400,000 cases, more than 80 per cent of the fruit is said to have been sold as damaged, at a total loss estimated at about £125,000, while the whole seaon's losses based on the prices of damaged fruit as compared with the

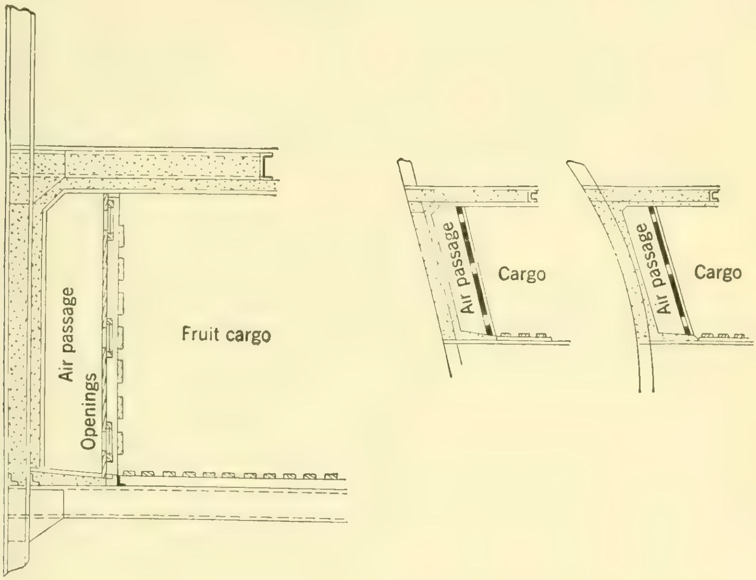


Fig. 314.—Marine Refrigeration—Ventilation.

market value for sound fruit at the same date, have been placed in the neighborhood of £250,000. The figures are taken from a statement prepared by the agent-general for Tasmania.

This type of damage, now know as brown heart, had previously been encountered in the Australian trade on a smaller scale. The nature and cause of the trouble were not, however, known, so that the matter appeared to be one urgently requiring scientific investigation.

The outstanding questions which it was hoped that the expedition would be able to answer conclusively were:

(*a*) Is the disease predetermined some time before its actual appearance, by "inherent vice"?

(*b*) Does brown heart appear before shipment, or in the holds during the voyage?

(*c*) What are the conditions either in the orchard, or during handling and transport which give rise to the disease, and how are they to be avoided?

The evidence obtained by the expedition, together with that already published, thus indicates that brown heart is developed on board ship during the homeward voyage, that the conditions which give rise to it are those accompanying insufficient ventilation of the hold, and that such conditions can be guarded against and all danger of the disease's occurring eliminated.

The Amount of Ventilation Necessary

The amount of ventilation necessary depends in the first place upon the rate of carbon dioxide formation in the holds. This in turn depends mainly upon:

(a) The weight of apples present;

(b) The inherent respiratory activity of the apples (varying in different seasons, at different stages of ripeness, and for different varieties);

(c) The temperature of the apples.

The second of these factors must be regarded for the present as an unknown variable, the values of which are, however, within known limits. We need only be concerned for the present purpose with the known maximum and minimum values.

In an ideal case, therefore, in which no accidental leakage is occurring, the amount of intentional ventilation by fans required is very roughly as follows:

Cargo, Temperature, Degrees F.	Percentage of Carbon Dioxide Maintained in the Hold	Ventilation, Cubic Feet of Air per Day per Ton of Apples
70	10	120
70	1	1200
35	10	25
35	1	250

In atmosphere control there are two alternatives. In the first place, safety—that is the avoidance of brown heart—may alone be aimed at. In that case, it would appear that holds with air circulation systems are on the whole satisfactory as at present operated. With the grid system, however, safety cannot be insured without gas-registering instruments and provision for ventilation.

The second and more scientific alternative is to maintain deliberately a desired carbon dioxide percentage level, with the purpose of taking advantage of the retarding effect of this gas on the ripening of the fruit. From this point of view, the amount of ventilation given by the air circulation systems would appear to be excessive, and the grid system without fans or trunks would seem to offer greater possibilities for atmosphere control. By restricting ventilation as far as is compatible with safety, the amount of snow on the grids is kept as low as possible, since ventilation implies the admission of outside air, and, unless this be artificially dried, it brings with it a large amount of moisture.

No definite statement regarding the amount of this intentional ventilation can be made to cover all cases, except that it should be such as to maintain the carbon dioxide concentration in the hold at not more than 10 per cent. In some cases, no

intentional ventilation may be necessary, the accidental leakage alone being sufficient. In other cases, it appears that as much as 10,000 cu. ft. of the hold atmosphere per day may have to be replaced by fresh air. The carbon dioxide percentage in the hold is the indicator by which the ship's engineer must regulate ventilation, and it would therefore be essential to have instruments of carbon dioxide in every hold. Atmosphere control without such instruments is as impossible as temperature control without thermometers.

Problem of Temperature Control

It is well known that the length of life of apples in storage is prolonged more or less proportionately by a reduction in temperature. This is due to two causes:

1. A retardation of the rate of ripening—the low temperature causing a slowing up of all vital processes.

2. The checking of mold development—the spores of these fungi, which are invariably present on the skin of the fruit, being largely prevented from germinating and invading the tissues.

The desiderata with regard to temperature are therefore:

1. Cooling of the fruit to 32 deg. F. to 34 deg. F. with the least possible delay after gathering.

2. The maintenance of a uniform temperature of 32 deg. F. to 34 deg. F. throughout the bulk of the cargo, and throughout the voyage.

The results obtained on the expedition show that it is necessary to recognize the great difficulties which have to be overcome in order to obtain even a fair working approximation to these desired conditions, and that the best means of overcoming these difficulties is not yet known. The difficulties increase in proportion to the bulk of the cargo carried, and the size of the hold employed. To-day, refrigerated cargo holds, carrying apples, may have a capacity of 80,000 cubic feet, and this space is filled with an almost solid block of apple cases.

Calculations based on data obtained by the expedition indicate that the amount of heat which has to be removed in cooling this cargo initially, say from 70 deg. F. to a carrying temperature of 35 deg. F., is about as great as the total amount of heat subsequently to be removed during a voyage of average length. The heat being generated at any time by the apples (due to respiration) is on the average three or four times as great as the heat leaking in through the insulated walls and floor from the sea. The heat which refrigeration has to remove therefore comes at all times largely from the cargo, which not only brings in a large amount of solar heat when loaded warm, but also continuously generates heat throughout its mass during the voyage.

The information required was obtained to some extent by the indirect method of constructing three-dimensional temperature maps of the hold, at different times, throughout the voyage. This method has the additional advantage of yielding direct results as to the extreme range of temperature distribution over the hold (in departure from uniformity), after the cargo has been cooled down, and has reached a steady state.

The results obtained by the expedition show that in large holds none of the systems in use attain a uniform temperature throughout the cargo after cooling. The temperature in different parts of the hold varies rather widely even after cooling down is completed and the steady condition of storage reached. There seems little doubt that slight modification in practice will effect considerable improvement in

this respect. Each of the different systems, however, presents its own particular technical problem—a problem of detail. In general, the most important agents in promoting both quick cooling and uniform temperature appear to be (1) roof grids and vertical dunnage or (2) high velocity, alternating, transverse, cold air draughts with transverse dunnage.

Chilled Meat.—Chilled meat is suspended from meat rails from the under side of the decks, and the best deck height in the clear is 6 ft. 6 in., although 9 ft. 6 in. can be used and even 12 ft. 6 in. or 15 ft. 6 in. below the rail has been successful, but extremes are bad on account of poor air circulation and the difficulty in stowing. Temperatures may be specified by the shipper, but $28\frac{1}{2}$ to $29\frac{1}{2}$ is a good value. As a rule, 35 days is the maximum storage for successful operation. Refrigeration is obtained by means of brine pipe coils suspended from under the ceiling and on

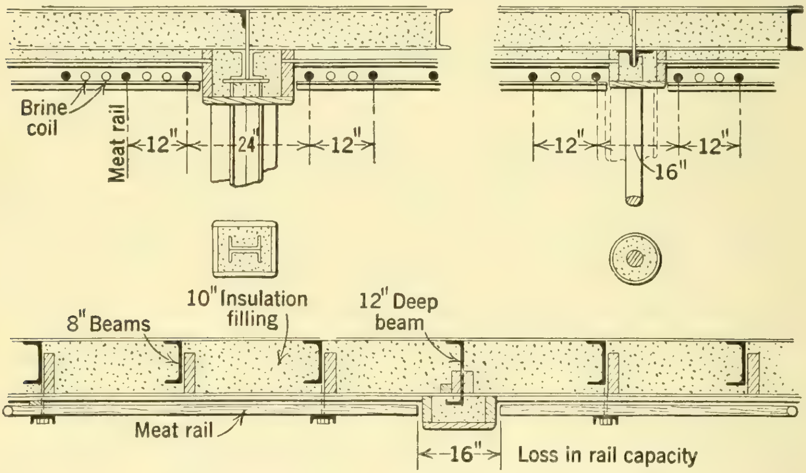


Fig. 315.—Marine Refrigeration—Piping, Insulation and Meat Rails.

the sides of the bulkheads, arranged preferably on the so-called interlaced system, with the cold, tepid and warm brine available with an attemperator (Fig. 318) for the proper control of the temperature. Too cold a temperature will cause a shrinkage of the meat, and usually a temperature of the brine of 15 to 18 deg. F. is good practice. The cargo is packed tight enough to prevent undue swaying and yet permit adequate air circulation. Bulkheads and side coils must be protected from the meat by means of battens. Figure 315 gives a good arrangement of rails and piping.

The General Carrier.—The general carrier must have space for all usual commodities, and be arranged to provide two temperatures.

Mechanical Equipment.—Lloyd's requirements specify sufficient tonnage to maintain the necessary temperature with 18 hours' operation. The capacity of the refrigerating machine using carbon dioxide while operating in the tropics is some 60 per cent of the capacity in the North Atlantic waters, whereas the ammonia machine is 80 per cent. Figure 316 gives an idea of the ratio of the storage space to the tonnage required, allowing for the reserve specified by Lloyd's, under nominal speed of the compressor. The kind of refrigerant used on shipboard is not mandatory. If ammonia is used it must be in a separate room and vented (usually on one of the upper decks), whereas carbon dioxide may be used in the main engine room. Some of the trade uses ammonia almost entirely, as, for example, the British ships in the River Plate

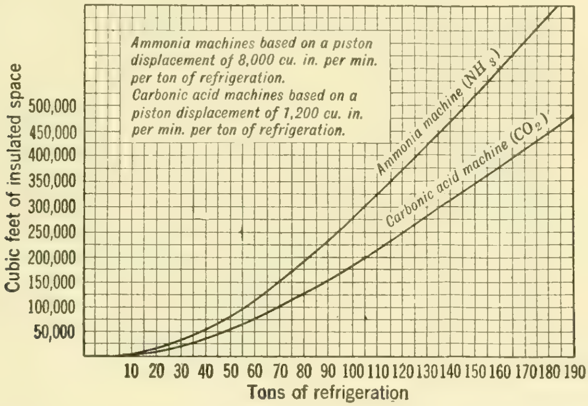


FIG. 316.—Marine Refrigeration—Tonnage Required of Carbonic and Ammonia Compressors.

service, and some of the largest refrigerator ships on the Australian and the New Zealand routes. The brine cooler room contains the brine tank, the distributing header and the pump connections. Insulation is usually of 4 in. corkboard in two layers, or its equivalent. The brine cooler is usually the double pipe or the shell and tube for ammonia, whereas with carbon dioxide the continuously welded iron pipe coils in separate nests in a cast iron box, or continuous circular iron pipe coils also nested are usual.

The condensers are of copper for carbon dioxide, of the submerged design, nested in rectangular boxes in the frame of the compressor. As copper cannot be used for ammonia, the double-pipe ammonia condenser as well as the shell and tube are mostly used. All equipment must be paired, or duplex, and sufficient spares must be carried in accordance with Lloyd's regulations.

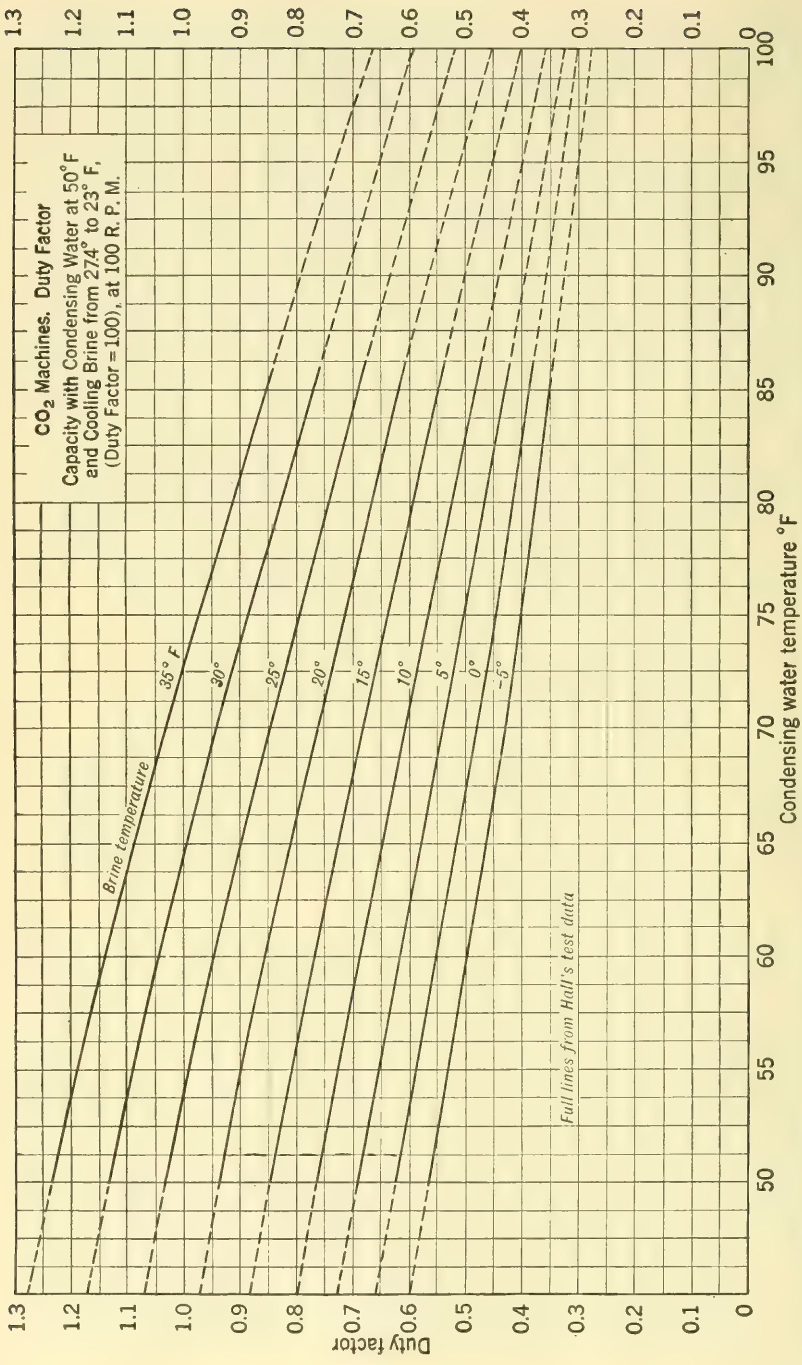


Fig. 317.—Effect of Condensing Water Temperature and Evaporating Temperature on Duty of Carbonic Compressors.

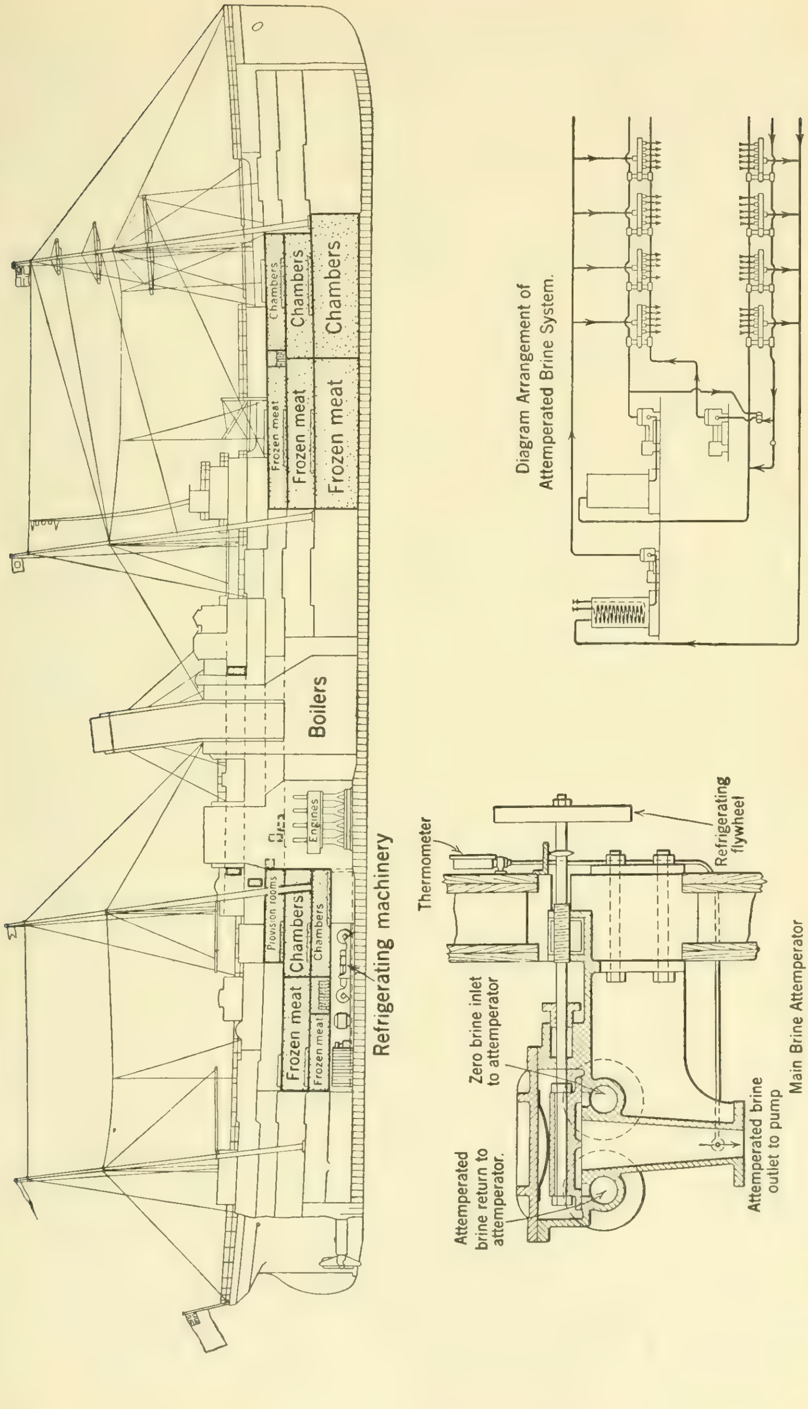


FIG. 318.—Marine Refrigeration Details.

DISTRICT COOLING (Pipe Line Refrigeration)

Pipe line refrigeration has, like central heating systems, particular and special applications. Its field is in the rather congested commission house and cold storage warehouse district, where the feed lines do not have to be made excessively long in order to pick up the load. The service needs to be fairly well established at the beginning as extensions cannot be made with economy after the initial installation, except to spread out into a wider area by installing a new compression plant at a new central location. The result has been that applications of pipe line refrigeration are relatively few in number and consist of plants located in the larger cities such as Boston, New York, Philadelphia, St. Louis, Kansas City, etc. The Quincy Market Cold Storage and Warehouse Company, of Boston, has changed from direct expansion to brine, and is now carrying over 1400 services with 1500 tons refrigeration of connected load, but the warehouse service is separate. The Merchants Refrigerating Company, of New York, has brine refrigeration only (at zero deg. F.), and carries 400 services from 200 to 15,000 cu. ft. of cold storage and besides has some small warehouses aggregating 2,750,000 cu. ft., of which 35 per cent is freezer and 65 per cent is cooler space (31 to 35 deg. F.). The St. Louis plant, however, is direct expansion, and in general it can be said that the brine system construction costs from 40 to 60 per cent more than does the ammonia direct expansion.

The choice as to whether a brine system is to be used or a direct expansion one, depends to a certain extent on local conditions and cannot be decided offhand without knowledge of all the facts. The pipe line system of refrigeration is in reality only a large ordinary plant where the mains are under the street. The construction cost is in favor of the direct expansion, especially when it is remembered that the liquid and the gas return lines to the compressor are not insulated. This is because of the fact that the evaporating coils in the cold storage boxes are very liberally piped, and the expansion valve is operated so that superheated ammonia is brought back to the street mains by having several boxes in series if necessary and thereby eliminating frost on the pipe lines in the street. The result of this procedure is that the return gas has the temperature of the pipe trench which usually in midsummer (as it is below the frost line) is at 70 to 75 deg. F., so that about 50 degrees of superheat is contained in the return gas. If direct expansion is employed the piping is usually designed for some 10 to 15 lb. per sq. in. drop from the most remote box to the compressor. If operation requires a boiling temperature corresponding to 25 lb. gage pressure, namely 12 deg. F., then approximately 10 lb. gage pressure will be had at the

compressor. The loss in capacity (as compared with 25 lb. gage and 10 degrees superheat) is 48 per cent, and the increased horse power per ton of refrigeration is 73 per cent. The loss of ammonia in the Boston plant before the change to brine operation is stated to have been as much as $15,000 to $20,000 per year.

When using brine both supply and return lines need to be carefully insulated. Judging from the experiences of the Boston and the New York systems, the larger plants are practically obliged to use brine. The control of the temperatures in the boxes is made possible very easily by the regulation of the amount of the brine flowing through the box coils, and there is no danger from the escaping fumes—should a break in the pipes occur—either to life or to the commodity being stored. With brine it is possible to install indicating and control systems, which will indicate by an alarm when leaks in the brine line develop, of the magnitude of approximately one-half barrel of brine within a determined short time interval, and on hearing this alarm the operator can close within 30 seconds all the main stop valves in the street system by closing the switch connected to the motors on these valves.[22] By this arrangement the brine system is divided into a number of parts, the leak is localized, and the remainder of the brine system can be put back on the load, without much loss of brine and without waiting for repairs to be made. Continuity of service is easier with brine than with direct expansion by utilizing brine storage, as, for example, the Merchants Refrigerating Company (operating with zero degree brine), has a large amount of strong calcium chloride brine kept at 32 deg. F., whereas the Boston plant circulates 10,000,000 gal. of brine per 24 hours and has 6,000,000 gal. in reserve. This plant makes use of a high-duty reciprocating pumping engine.

As regards the cost of the operation of the brine system, this is, of course, dependent on the design. The problem is very similar to that of a cold storage warehouse. The circulating pump may have only the friction head and the velocity head to overcome in the case of balanced systems, and it may have in addition the lift to the highest point in the piping if the system is an open one. The pressures against which the pump has to operate are given by A. W. Oakley as 60.5, 55.5, and 50.5 lb. per sq. in., corresponding to 113.5, 104.2, and 95.0 ft. of brine head respectively. Using a nominal value of 6.0 gal. of brine per ton of refrigeration per minute, and a density of 1.23, the horse power per ton becomes

$$\frac{6.0 \times 1.23 \times 8.33 \times 113.5}{33{,}000} = 0.211.$$

[22] A. W. Oakley, Refrigerating Engineering, December, 1924.

Piping.—The piping for brine can be full weight steel or wrought-iron pipe, using flanged joints and ring gaskets and in some cases the alternate joint is made with heavy cast-iron sleeves and is caulked with lead, or this may be of cast iron using the bell and spigot, the flanged and screw thread or the gland end types. For ammonia it is usual to employ full weight pipe for pipe sizes 4 in. and over and extra heavy pipe for sizes under 4 in. The Merchants Refrigerating Company uses a dual system, so that if one circuit develops a leak the other circuit can continue with the load until the leak is repaired. The direct-expansion ammonia system requires besides the liquid and the suction gas line a third line for pumping out purposes. It is necessary to place all piping below the frost line (from 3 to 7 ft. below the surface), and if possible to confine all connections to the manholes which are located at the intersecting streets. The tendency seems to be to use all welded pipe as far as possible between manholes. Expansion joints must be provided every 500 ft. at a maximum, and every 300 ft. where heavy distributing pipe lines are taken off the mains. These expansion joints should be located at the manholes, and they may be of the slip joint, the corrugated or the U-type, the pipe being suitably anchored at some convenient point thereby permitting the change of length of the pipe to be taken up by the expansion joint. The pipe is subjected to a possible change of temperature of from 70 to 100 deg. F. The change of pipe length corresponding to this temperature change is 0.75 in. per 100 ft. of pipe. If the slip joint type is used the finished surface of the sleeve should be kept smeared with grease so as to prevent tearing of the packing of the gland which may under extreme conditions have a movement of of 2.5 to 3.0 in. Valves not placed in the manholes should have an extension valve stem. Service branches should not come up into the frost lines, and should take off from the top of the main, using a nipple and ell so as to permit some freedom of movement to conform to the expansion and the contraction of the main. The coils in the boxes should all have stop valves in the supply and the return, and (for brine) air valves at the high points and a lock valve so as to regulate the maximum amount of brine flow. The brine cooler generally used is the shell and tube design, since this is efficient and easily cleaned. The number of passes of the brine varies considerably, and there may be two or more up to (possibly) eight, but as a rule four passes of the brine through the tubes are sufficient.

Insulation.—The type of insulation varies with the different installations. In Fig. 319 there are several methods in use. If a wooden box is used the wood should be waterproofed (kananized treated). The pipe conduit or box is packed, frequently, with a concrete of granulated cork and pitch.

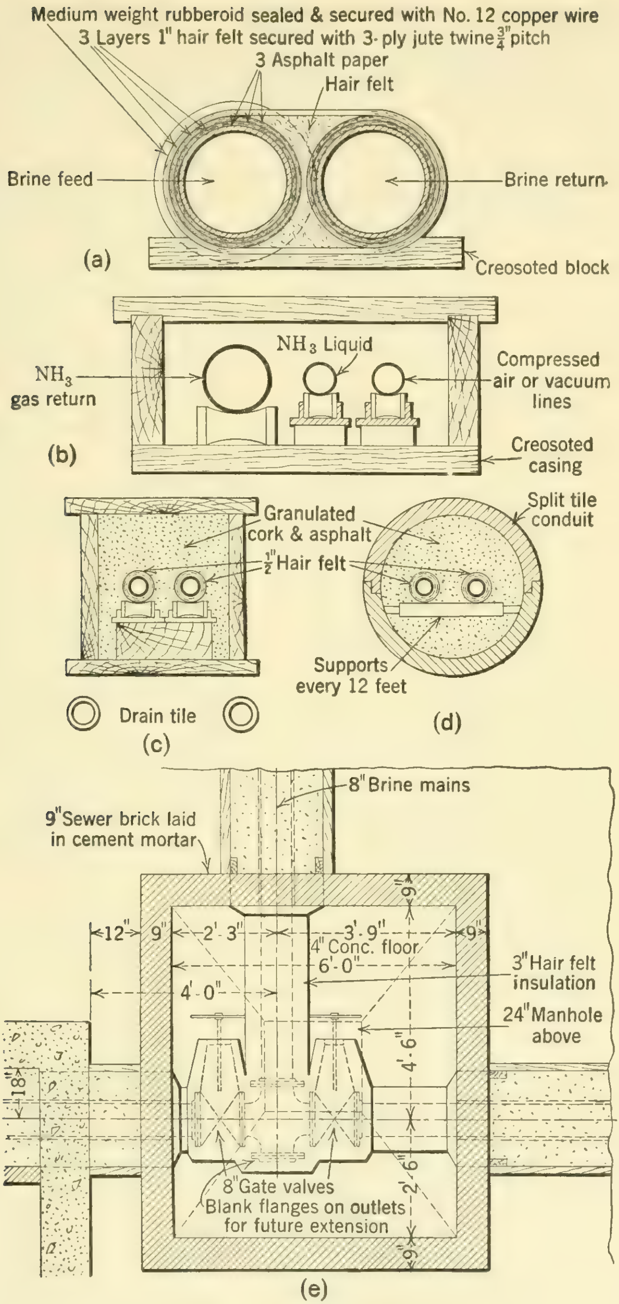


FIG. 319.—District Cooling Details of Piping.

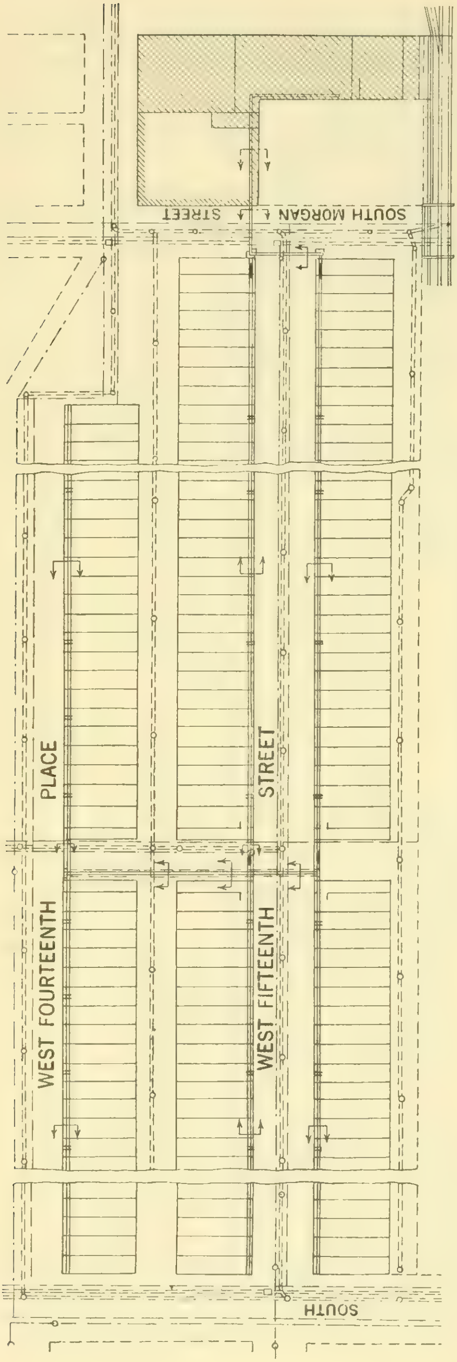


FIG. 320.—District Cooling—Street Piping.

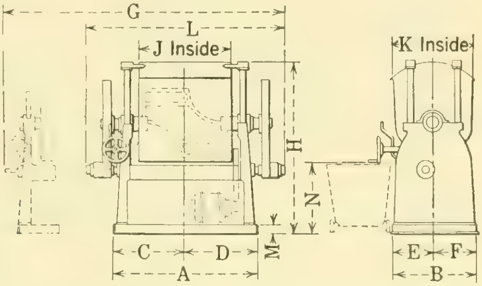
R. H. Tait[23] says that in St. Louis 59 per cent of all connections are of 300 cu. ft. capacity or less, 17 per cent are of 300 to 600 cu. ft., 14 per cent are of 600 to 1200 cu. ft., and 10 per cent are of 1200 and over, and that he figured on 7500 cu. ft. of cold storage at 32 to 36 deg. F. per ton of refrigeration, whereas the practice in Boston, according to F. L. Fairbanks, is only 2800 cu. ft. The load on the plant depends on the relative amounts of freezer and cooler space, as the freezer is refrigerated for from 9 to 12 months, whereas the cooler is under load only about 6 months. The larger boxes are usually charged according to the refrigeration supplied, but the smaller ones are on a flat rate per cubic foot, depending on the details of the box and the temperature carried in the room.

REFRIGERATION IN THE BAKERY

The modern bakery of 100,000 loaves per week includes in its process

[23] R. H. Tait, American Society of Refrigerating Engineers, 1913.

several steps where mechanical refrigeration is now considered an economic necessity. These include (*a*) the 50 deg. F. cold storage room of about 100 to 200 sq. ft. floor area for yeast, shortening, malt and milk; (*b*) a cooling tank for lowering the temperature of the dough water to 40 degrees, or lower, on the basis of about 25 per cent water by weight; (*c*) the cooling of the mixers either by the use of brine at 10 to 20 deg. F. or water coils with water at about 35 degrees with the addition of crushed ice. The water coil system is the popular method of cooling, but blowing the air at about 32 degrees is increasing in use; (*d*) cool the air in the dough room (mixer) down to 78 degrees when this is necessary and (*e*) provide a cold storage for waxed paper at 55 to 60 deg. F. to prevent the paper sticking.[24]



BBL	A	B	C	D	E	F	G	H	J	K	L	M	N	H.P.	CLUTCH SHAFT SPEED
$1\frac{1}{2}$	51"	32"	$25\frac{1}{2}$"	$25\frac{1}{2}$"	16"	16"	5' 9"	5' $8\frac{3}{8}$"	27"	26"	6'-3"	$4\frac{1}{4}$"	$28\frac{1}{2}$"	8	243
2	51"	32"	$25\frac{1}{2}$"	$25\frac{1}{2}$"	16"	16"	5'-9"	5' $8\frac{3}{8}$"	$27\frac{3}{8}$"	$26\frac{3}{4}$"	6'-3"	$4\frac{1}{4}$"	$28\frac{1}{2}$"	3	243
$2\frac{1}{2}$	58"	35"	29"	29"	$17\frac{1}{2}$"	$17\frac{1}{2}$"	6' 4"	5'-11"	38"	31"	6'-10"	$4\frac{1}{4}$"	$28\frac{1}{2}$"	5	236
3	58"	35"	29"	29"	$17\frac{1}{2}$"	$17\frac{1}{2}$"	6'-4"	5'-11"	34"	32"	6'-10"	$4\frac{1}{4}$"	$28\frac{1}{2}$"	5	236

FIG. 321.—Bakery Refrigeration—Dough Mixer.

In mixing the dough it is desired to develop the gluten so that the structure of the dough will be sufficiently elastic and strong enough to hold in confinement the gas which is produced by the fermentation process. The power which is consumed in the mixing develops heat (the heat equivalent of the work done) and the more rapid the mix the faster the temperature rise will take place. A certain type of mixer requires 20 minutes for the mix, and every minute over 10 minutes gives a marked increase in the quality, but should heating prevent the last part of the process an inferior bread will result. The cooling by blowing cold air into the dough as it is mixed has a bleaching effect and produces a white bread, and it appears to assist fermentation.[25]

[24] W. W. Reece, Chicago Section, A.S.R.E., October, 1923.
[25] H. L. Fisher, A.S.R.E., April, 1923.

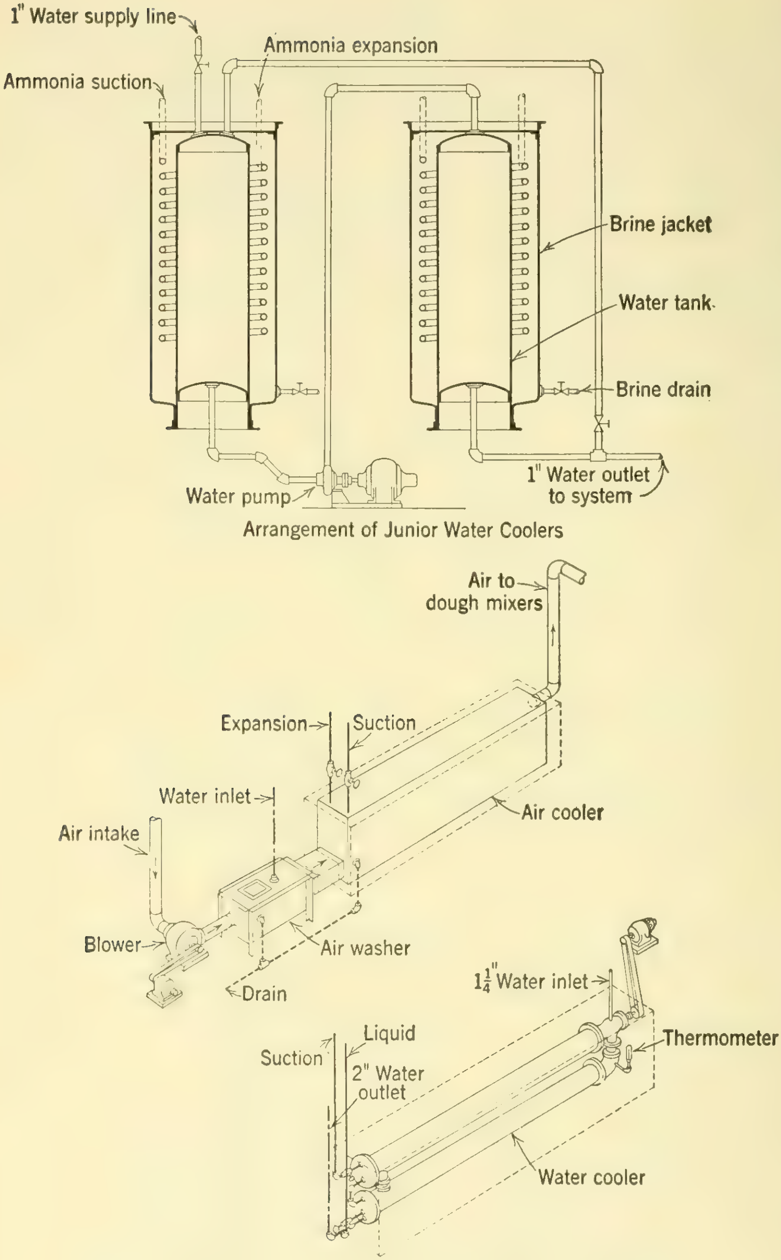


FIG. 322.—Bakery Refrigeration—Water and Air Coolers.

In bakery refrigeration it is usual to figure on 250 loaves of bread per barrel of flour of 196 lb. The dough mixers vary in size (Fig. 320), but the 3-barrel type is the one which operates at greatest advantage Such a size would use 588 lb. of flour and 110 lb. of water, cooled to 35 degrees. During mixing, 750 cu. ft. per min. of air at 32 deg. F. and under $3\frac{1}{2}$ oz. pressure is supplied each mixer. The problem of the calculation of the refrigeration requirements varies with the location (as regards the maximum summer temperatures) and on the construction of the building, the exposure, the window areas and other pertinent

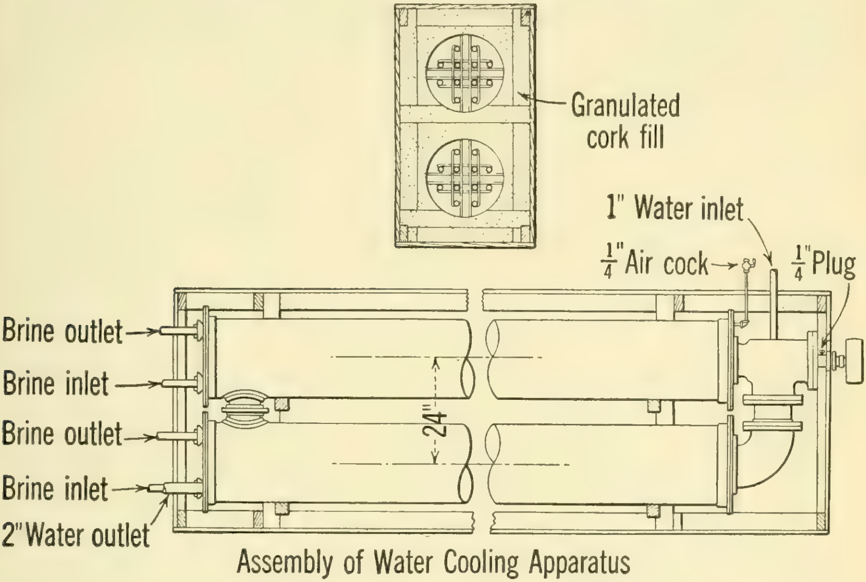


FIG. 323.—Bakery Refrigeration—Water Coolers.

factors. As an illustration, the following will give an idea of the larger refrigerating quantities:

Problem.—Find the refrigeration required in a bakery of 60,000 loaves capacity per 8 hours per day. The outside air temperature in 90 and the wet bulb is 75 deg. F.

The capacity is $60{,}000 \div 250 = 240$ bbl. per day = 30 bbl. per hr. At the rate of 20 min. per batch there will be an average of $30 \div 9 = 3\frac{1}{3}$ mixers in continuous service. The air requirements will be: $3\frac{1}{3} \times 750 = 2500$ cu. ft. per minute to be cooled from a wet bulb temperature of 75 degrees to 32 degrees. The volume of 1 lb. of air is 12.47.

$$2500 \div 12.47 \times (37.8 - 11.8)^{26} = 5220 \text{ B.t.u.}$$

[26] In all air cooling the psychrometric chart (Fig. 326) may be used.

Assuming that the initial water temperature is 80 deg. F., the heat removed is:

$$\frac{110 \times 3\frac{1}{3} \times (80 - 35)}{20} = 825 \text{ B.t.u.}$$

$$\text{Total refrigeration} = 6045 \text{ B.t.u.}$$
$$= 30.2 \text{ tons.}$$

As the preceding calculation is liberal it will provide for the small requirements of the storage rooms for the waxed paper, the yeast, shortening, malt and the milk. In fact, plants of this capacity in and near the vicinity of Milwaukee use about 20 tons of refrigeration, but possibly with a longer day than eight hours, in which case the calculated capacity would be reduced in proportion. Figures 321, 322 and 323 give details of bakery refrigeration.

CHAPTER XV

THE COOLING AND CONDITIONING OF AIR

One of the most important applications of mechanical refrigeration is that of the cooling and conditioning of air, which, being a fluid, can be made to act conveniently as a carrier of refrigeration. Examples of air cooling and conditioning are: in the process of the chilling of glue, cooling films for cameras and for motion pictures, or even for making gelatine capsules, either because the emulsion needs to be hardened quickly or because the commodity must be chilled quickly from the fluid to the semi-fluid or solid condition. At times the problem becomes one of simply keeping the workroom moderately cool and properly humid, or it may be (as in certain chemical plants) that of lowering in temperature sufficiently the containers of a gas being handled to prevent the danger of its escape into the atmosphere of the workroom. An example of this last is to be found in the case of the shell-filling stations where phosgene gas is charged into shells. Just what system of air cooling needs to be employed in each case depends on the particular process. In extreme cases what is required particularly is a certain low humidity of the air, obtained by cooling the air down to a temperature sufficient to give the desired water content to the air. Such a condition the Gayley method, now obsolete, tried to meet by condensing out the water content of the air by cooling it down to 25 deg. F. (it being desired to hold the moisture content at 20 grains of water per one pound of dry air), subsequently heating it to the usual blast temperature of 3000 deg. F. before supplying it to the blast furnace.

In cooling air it is possible to use a bunker room through which the air is forced and brought into contact with the refrigerated pipes, but this method is poor because of the likelihood of heavy frostation on the pipes and the consequent lowering of the efficiency of heat transfer. The more usual method at present is to cool by the use of a spray chamber, using a water spray in the case of temperatures above 35 deg. F., and water and brine sprays in two separate stages in cases of temperatures below 35 degrees. When water is used it is usually cooled by the use of a Baudalot cooler (Fig. 342), where the refrigerant boils in the inside of the pipes and the water trickles from the outside of the top pipe down to the lowest pipes by gravity. The air to be cooled is circulated through an air chamber where it comes into intimate contact

with the water which is atomized by the nozzles through which the water passes.

Moisture Content.—In air cooling and conditioning certain terms, peculiar to the conditions, are used. The *wet bulb* temperature is that temperature obtained when a thermometer with a wetted gauze is exposed to the air in motion—preferably at a velocity of 15 ft. per second. It is a depressed temperature, below that of the ordinary dry bulb temperature, and the amount of this depression depends on the moisture content in the air. The maximum amount of water content which can be absorbed by a unit weight of dry air at constant pressure depends on the air temperature, and this varies from 44.2 grains at 45 deg. F. to

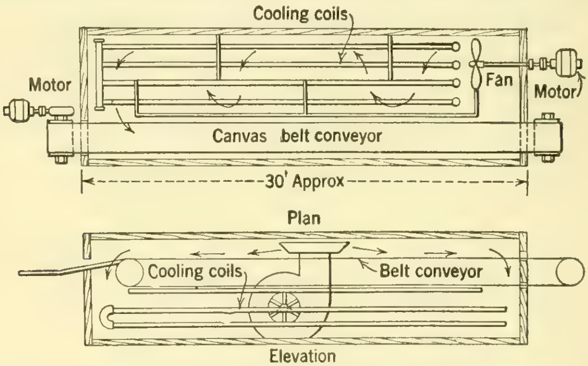


FIG. 324.—Method Employed in Cooling Certain Commodities.

155.8 grains at 80 deg. F. per 1 lb. of dry air. The "*dew point*" temperature is that temperature at which air is saturated with moisture and beyond which any further reduction of temperature will cause a precipitation of water.[1] Incidentally, the dew point and the wet bulb temper-

[1] The humidity may be expressed approximately as a ratio of the pressures. If p_1 is the pressure of the water vapor at some given condition, p_2 the pressure of saturation at a lower temperature, and p_3 the pressure of saturated vapor at the temperature corresponding to the given condition p_1, it can be stated approximately that

$$p_1 = p_2 = MBT;$$
$$p_3 = M_1BT;$$

therefore

$$\frac{M}{M_1} = \frac{p_2}{p_3},$$

where $V_1 = 1.0$;
M = the weight in lb.;
B = the gas constant;
T = the absolute temperature in deg. F. abs.

or the humidity is the ratio of the dew point pressure to the pressure of saturation at the temperature of the given condition.

atures are quite distinct. The *relative humidity* can be defined as the ratio of the weight of the moisture present in a certain volume of air to the amount that could be present if the air were saturated. This value is usually expressed as a per cent.

The Psychrometric Chart.—Fig. 325[2] represents a chart used for solving problems in air cooling. In the diagram it will be noticed that such a point as "*A*" represents a condition of 65 deg. F. dry bulb temperature and 60 per cent relative humidity. Lines sloping downward to the right are lines of constant wet bulb temperature, and "*C*" is the temperature (56.7 deg. F.) of the wet bulb and "*B*" is the dew point temperature (50.7 deg. F.). The moisture content of "*A*" is 55 grains per one pound of dry air, as is every other point on a horizontal line drawn through "*A*." In the diagram the *total heat* measured from 0 deg. F. is given by the curve (marked "total heat"). It has been found by experiment that the heat removed in the cooling of air can be obtained by getting the difference in the total heats correspond to the wet bulb temperature at the beginning and the final conditions. For example, if air is to be cooled from a condition of 65 deg. F. dry bulb and 60 per cent relative humidity to the dew point temperature, the total heats from the diagram are 24.1 and 20.75 B.t.u. respectively. The volume of a pound of dry air with various moisture contents can be found by referring to the two volume curves; for example, at 51 deg. F. the volume of one pound of dry air is 12.87 cu. ft., and when saturated with moisture it is 13.03 cu. ft. With the foregoing it is now possible to proceed with an illustrative example, and this will be made general enough so that the method of solution may be used in other problems if the details of the particular job are known. In calculating the refrigeration required it is generally necessary to make allowance for all sources of heat, i.e., the heat equivalent of the work done, the heating effect of the electrical losses (the I^2r losses), all process heating, the heat generated by the illumination, animal heat, heat of chemical reactions, etc.

Problem.—A room in an industrial plant is to be maintained at a temperature of 65 deg. F. and 60 per cent relative humidity (Fig. 327). Its cubical contents is 204,300 cu. ft., and it has 1248 sq. ft. of glass and 908 sq. ft. of wall surface exposed to the sun's rays, and 2154 sq. ft. of glass, 1863 sq. ft. of wall, 2728 sq. ft. of partition and 12,770 sq. ft. of floor area subject to heat leakage. The outside temperature of the air will be taken as 95 deg. F. maximum, with a wet bulb temperature of 76 deg. F. The infiltration loss will be assumed as a complete change of the air of the room in five hours. The maximum number of operators will be taken as 500 persons at any one time, and the illumination will be taken as 39 250-watt lamps. It is required to find the total refrigeration, the size of the fan required to

[2] Willis Carrier, Transactions of the American Society of Mechanical Engineers, 1911.

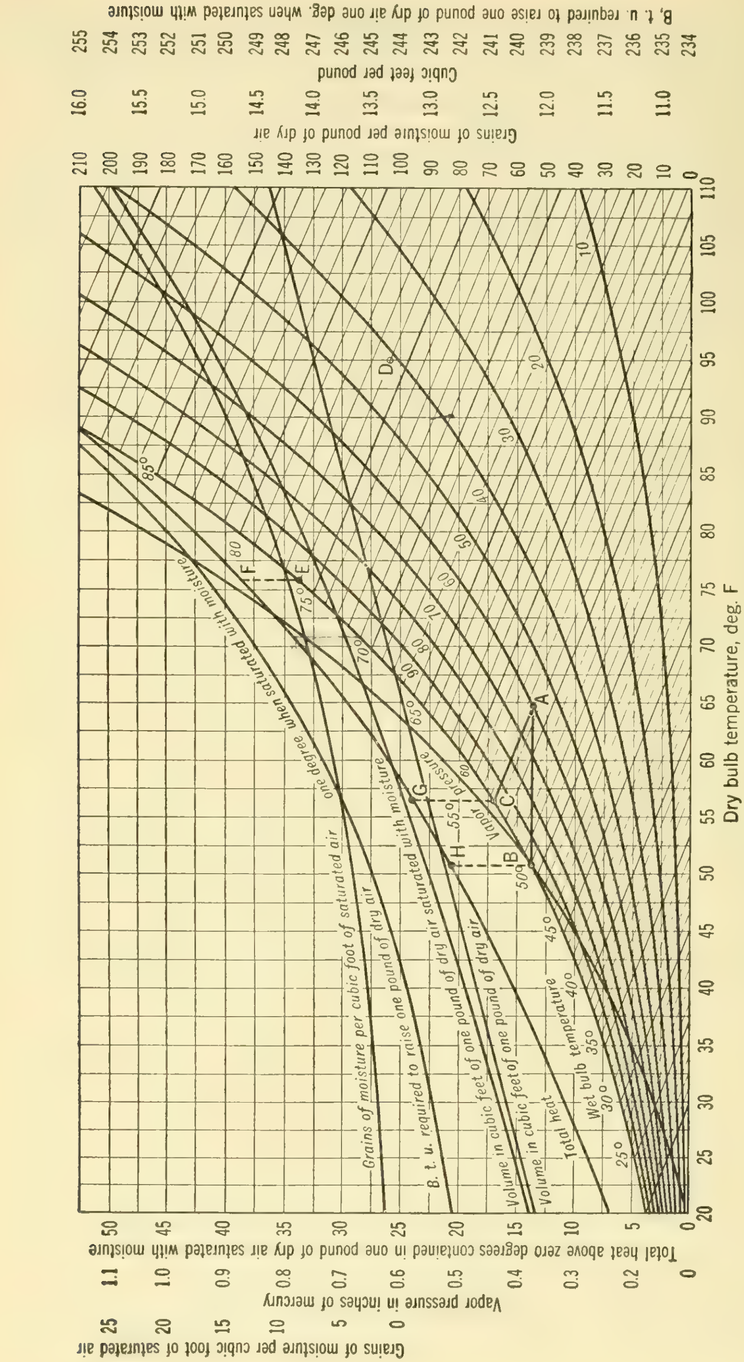


FIG. 325.—The Psychrometric Chart.—Method of Using.

circulate the air and the amount of water required to be pumped through the sprays, if the water is to be heated by the air from 43 to 51 deg. F. Also it is required to find the number of sprays, the amount of baffles and eliminator surface, and the size of the ducts to convey the cooled air to the different parts of the room.

Calculation.—Referring to Fig. 325, point A is found at the intersection of the 60 per cent humidity line and the dry bulb temperature of 65 deg. F. The line AB is drawn so as to show the process of heating or cooling at a constant water content in the air—in this case a constant weight of 55 grains of water per pound of dry air. Point B indicates that the dew point required from the conditions of the problem must be 50.7 deg. F., and the wet bulb temperature (point C) must be 56.7 degrees. If the air passed through the spray chamber is to have the *sensible* heat only removed from it so that the moisture content will always remain at 55 grains, then the air

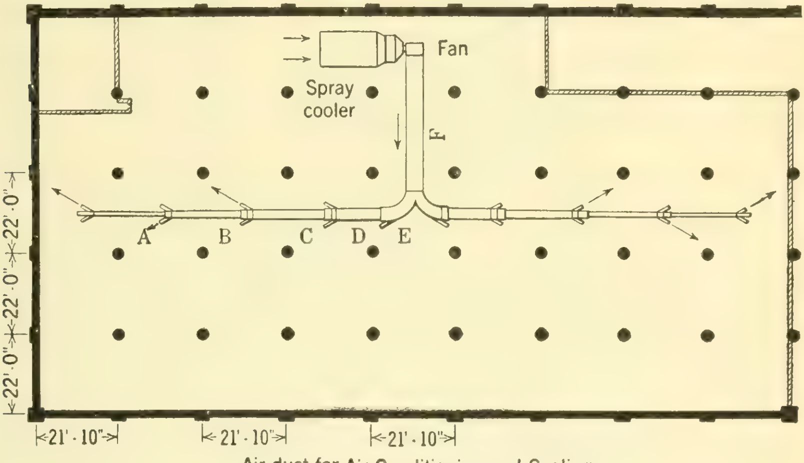


FIG. 327.—Problem in Air Conditioning.

must be cooled down to 50.7 deg. F. and sufficient air must be circulated in order that the total heat entering the room, and tending to raise the air temperature, will be removed by the sprays.

Heat Leakage.—The heat leakage will be:

The effect of the direct radiation (of the sun):

1248 sq. ft. of glass at 0.5 B.t.u. per sq. ft. (per minute) = 624 B.t.u. per minute.
908 sq. ft. of wall at 0.1 B.t.u. per sq. ft. (per minute) = 91 B.t.u. per minute.

(These values of 0.5 and 0.1 are practical experimental values to account for the radiant heat of the sun's rays.)

People. $500 \times 5.0 = 2500$ B.t.u.

(The value 5.0 is the *sensible* heat given off by each workman per minute. This and the other values in the calculation of the amount of air to be circulated by the fan are the heat units tending to raise the temperature of the air.)

Heat Leakage (by conduction) [3]:

Glass.	$2{,}152 \times 0.46 \times (95 - 65)$	= 29,700 B.t.u. per hour.
Wall.	$1{,}863 \times 0.075 \times (95 - 65)$	= 4,200 B.t.u. per hour.
Partition.	$2{,}728 \times 0.09 \times (95 - 65)$	= 7,370 B.t.u. per hour.
Floor.	$12{,}770 \times 0.08 \times (95 - 65)$	= 30,700 B.t.u. per hour.
	Total heat leakage per hour	= 71,970 B.t.u.
	Total heat leakage per minute	= 1,200 B.t.u.

Infiltration. $$\frac{204{,}300 \times 0.250 \times (95 - 65)}{60 \times 5.0 \times 13.5} = 379 \text{ B.t.u. per minute.}$$

Illumination. $$\frac{39 \times 250 \times 42.4}{746} = 556 \text{ B.t.u. per minute.}$$

Total sensible heat per minute	= 5,350 B.t.u.
5 per cent for contingencies	= 267 B.t.u.
Estimated heating effect of the circulating fan	= 272 B.t.u.
Total	= 5,889 B.t.u. per minute.

This amount of sensible heat, 5889 B.t.u., is the total heat entering the room from all the sources tending to raise the temperature of the air in the room, and this is the only factor affecting the capacity of the fan. As refrigeration is to be accomplished by means of water sprays in a spray chamber it is necessary to move *enough* air through this chamber so that heat will be absorbed by the water at the rate of 5883 B.t.u. per minute. This air so circulated leaves the room at 65 deg. F. and reenters it again from the sprays at 50.7 deg. F. In other words there is a rise of temperature of 14.3 degrees, and the ability of the air to absorb heat will depend on this rise of temperature and its specific heat. If the specific heat is taken as 0.245 B.t.u., then the number of pounds of air to be circulated per minute will be

$$\frac{5889}{14.3 \times 0.245} = 1680 \text{ lb.}$$

As the air entering the fan is saturated with moisture the specific volume will be 13.03 cu. ft., and the total volume required at the entrance of the fan will be

$$1680 \times 13.03 = 21{,}890 \text{ cu. ft. per minute.}$$

Refrigeration Required.—The preceding calculation was only for the purpose of estimating the size of the fan. The heat absorbed by the air is *not* the refrigeration required of the refrigeration plant. As a matter of fact it is only about one-half of the required amount. In the present problem a certain amount of air has to be renewed, and a certain amount may be recirculated. In this problem 10 per cent will be fresh air, and 90 per cent will be recirculated, and will be cooled from the conditions of 60 per cent humidity and 65 deg. F. dry bulb to 50.7 degrees. The fresh air will be cooled from 95 deg. F. dry bulb and 76 wet bulb to the same temperature of 50.7 degrees. The total heat in either case will be found by going upwards

[3] These values can be found by the method given in Chapter VI.

from the wet bulb temperature, 100 per cent humidity line, to the total heat line and then reading the total heat from the corresponding point on the scale.

The total heat at 76 deg. wet bulb	38.7 B.t.u.
The total heat at 50.7 deg. wet bulb	20.75 B.t.u.
Heat removed per 1 lb. air	17.95 B.t.u.

The total cooling of the fresh air will be $0.10 \times 1680 \times 17.95 = 3020$ B.t.u.

The total heat of the air at 56.7 deg. F. wet bulb	24.20 B.t.u.
The total heat of the air at 50.7 deg. F. wet bulb	20.75 B.t.u.
Heat removed per 1 lb. of air	3.45 B.t.u.

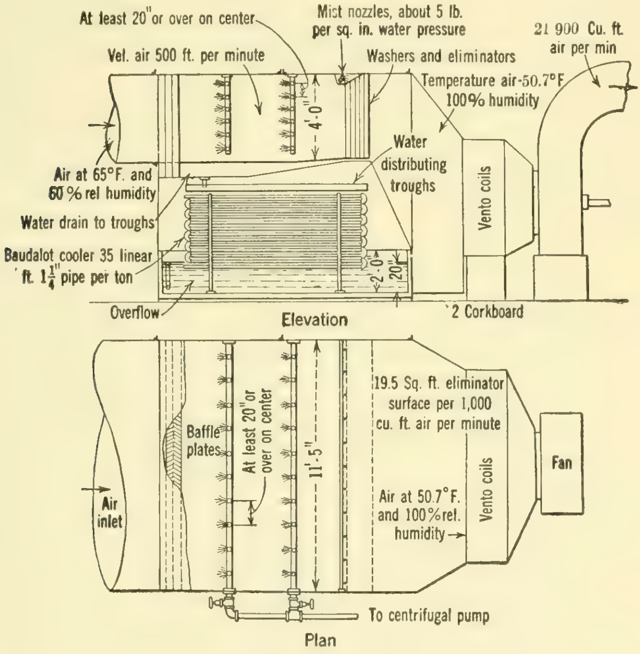


FIG. 328.—Water Spray Chamber for Air Conditioning.

The total cooling of the recirculated air is $0.90 \times 1680 \times 3.45 = 5210$ B.t.u. A certain amount of moisture will enter the air in the room, which will tend to increase the moisture content. This will be condensed in the spray chamber, and will add this much more to the refrigerating load. The amount of this moisture entering with the air of infiltration is

$$\frac{204{,}300}{5 \times 60} = 681 \text{ cu. ft. per minute.}$$

The volume of 1 lb. of air at 95 deg. F. and 42 per cent relative humidity is equal to $13.98 + (0.82 \times 0.42) = 14.32$ cu. ft. When this air enters the room it will be

cooled to 65 degrees and some of the water content will be condensed. The excess moisture brought into the room will be then:

$$\frac{681 \times (108 - 55)}{14.32 \times 7000} = 0.360 \text{ lb.}$$

As only 90 per cent of the air in the room is recirculated the refrigeration required is

	$0.90 \times 0.360 \times 1060 =$	344 B.t.u.
To these must be added:		
The latent heat of the operators,	$0.9 \times 500 \times 3.0 =$	1,350 B.t.u.
The heat effect of the water circulating pump (estimated) at 5 hp.	=	212 B.t.u.
Radiation effect into the room (estimated)	=	50 B.t.u.
Total refrigeration	=	10,186 B.t.u.
5 per cent for safety	=	509 B.t.u.
		10,695 B.t.u.
		or 53.50 tons

The water in the spray chamber is heated from 43 to 51 degrees and the number of pounds of water required will be

$$10{,}695 \div (8.3 \times 8.0) = 161 \text{ gallons per minute,}$$

and the circulating system should be designed for at least 180 gallons per minute so as to carry any overload for a short time should occasion demand it.

The Spray Chamber.—Having found the refrigerating load for the room to be 53.5 tons, the amount of air to be cooled to 50.7 deg. F. and circulated to be 21,900 cu. ft. per minute and the amount of water to be pumped through the sprays to satisfy the conditions of the problem to be 180 gal. per minute, the next step is to determine the amount of surface required in the spray chamber and the sizes of the various pieces of apparatus to be used in the spray chamber. First, however, it should be said that with the temperatures required in this type of problem water is always used for cooling the air, and a form of spray nozzle is used that will atomize the water. Cooling coils in the spray chamber may or may not be used (kept wet by means of flooding nozzles) in order to assist in the cooling of the air. In this design the sprays only will be used, and the eliminator plates (provided to remove the surplus water carried in the air) will be provided with special flooding nozzles in order to keep them clean, and to increase their efficiency. The smaller air cooling plants may have a Baudalot water cooler directly underneath the spray chamber so arranged as to be flooded by the water passing out of the spray chamber, in a manner similar to the method used with atmospheric condensers receiving water from cooling towers and spray nozzles. The objection to this method where ammonia is used is that there is no particular safety from bad leaks, although the ammonia passing out through a small leak would be absorbed by the water for a time and there would be some warning of the fact of the leak. This method will be used in the present design on account of its simplicity, but it is well to keep in mind that the use of brine might easily be imperative at times. However, with the use of extra heavy pipe, first class fittings and reasonable care and inspection during erection and operation there should not be any appreciable danger (Fig. 328).

Number of Spray Nozzles.—Referring to the table of the capacities of spray nozzles (Table 101), it will be seen that with the $\frac{3}{16}$-in. diameter nozzle opening and a pressure of 20 lb. per sq. in., 15.0 lb. of water per minute will be sprayed, or $15 \div 8.3 = 1.8$ gal. per minute. The problem requires 161 gal. per minute at the very minimum, and 180 gal. should be allowed for. There will be required then:

$$180 \div 1.8 = 100 \text{ nozzles under a pressure of 20 lb. or}$$

$$180 \div \frac{17}{8.3} = 88 \text{ nozzles under a pressure of 25 lb.}$$

These will be arranged in two banks, each of 50 nozzles, spaced not more than 20 in. apart, and not less than 6 in. between centers in order that a satisfactory "curtain" of cold water is provided for the air to pass through. As the velocity of the air in this part of the system must be not more than 500 ft. per minute, the cross-sectional area becomes $22{,}000 \div 500 = 44.0$ sq. ft. If the head room limits the height of the chamber proper to 4 ft. 0 in., then the width becomes 11ft. 0 in. Assuming that the nozzles are to be 12 in. on centers horizontally, then 10 stands will be required of 5 nozzles high. Evidently some arrangement, say using 10 stands spaced 13 in. on centers horizontally and 5 nozzles high on $7\frac{1}{2}$ in. centers will give satisfactory spacing.

TABLE 101

Spray Nozzles

Capacities in Pounds of Water Discharged per Minute under Varying Pressures
(Carrier Engineering Corporation)

Diameter of Pipe Connections, Inches	Diameter of Openings, Cap. Inches	Pressures in Pounds per Square Inch										
		10	20	25	30	40	50	60	70	80	90	100
$\frac{1}{4}$	$\frac{1}{32}$	.27	.39	.44	.48	.55	.62	.68	.73	.79	.83	.89
$\frac{1}{4}$	$\frac{1}{16}$	1.0	1.5	1.7	1.8	2.1	2.4	2.6	2.8	2.9	3.2	3.4
$\frac{1}{4}$	$\frac{3}{32}$	2.3	3.3	3.7	4.0	4.6	5.2	5.7	6.1	6.5	6.9	7.3
$\frac{3}{8}$	$\frac{1}{8}$	4.6	6.5	7.3	8.0	9.2	10.0	11.0	12.0	13.0	14.0	15.0
$\frac{3}{8}$	$\frac{3}{16}$	10.0	15.0	17.0	18.0	21.0	23.0	26.0	28.0	29.0	31.0	33.0
$\frac{1}{2}$	$\frac{1}{4}$	18.0	26.0	29.0	32.0	37.0	41.0	45.0	49.0	52.0	56.0	58.0
$\frac{3}{4}$	$\frac{3}{8}$	51.0	72.0	81.0	88.0	102.0	114.0	125.0	135.0	144.0	153.0	162.0
1	$\frac{1}{2}$	91.0	128.0	144.0	157.0	181.0	202.0	222.0	240.0	257.0	272.0	
$1\frac{1}{4}$	$\frac{5}{8}$	127.0	182.0	204.0	226.0	257.0	287.0	310.0				
$1\frac{1}{2}$	$\frac{3}{4}$	186.0	263.0	295.0	332.0	370.0	415.0					
2	$\frac{7}{8}$	252.0	356.0	400.0	435.0	501.0						
2	1	330.0	465.0	522.0	570.0	657.0						
$2\frac{1}{2}$	$1\frac{1}{4}$	515.0	728.0	815.0	890.0							

1 Cu. ft. of water at 4 deg. C. weighs 62.428 lb.
1 gal. of water weighs 8.3 lb. = 231 cu. in.

Eliminators.—If the baffle plate construction is used in order to reduce the eddy currents, give a uniform velocity to the air, and prevent water from splashing out

of the spray chamber, it is customary to use just enough to rectify the air (Fig. 335). The eliminator surface is calculated on the basis of $19\frac{1}{2}$ sq. ft. (calculated for one side only) of surface per 1000 cu. ft. of air per minute or $(22{,}000 \div 1000) \times 19.5 =$ 430 sq. ft. of surface is wanted. The *Carrier* eliminator construction is one with crimped gutters set vertically on $1\frac{1}{8}$-in. centers, each with six corrugations at 60-degree directional change of the air, each corrugation about $2\frac{5}{8}$ in. wide, and arranged so that the first three will have smooth sides and corners and the last three will have stamped lips for catching the entrained moisture.

The *Webster* eliminator is designed for four corrugations, with 30-degree deflection of the air, made up of two plates: the rear staggered with respect to the front, the separate plates spaced about 3 in. on centers, and 5 in. wide. The Webster practice is to discharge the water spray in the direction of the air flow, and to place them not less than 4 ft. 3 in. in front of the eliminators. These sprays use about 20 lb. per sq. in. pressure at the nozzle, and in addition "mist" nozzles are used at 5 lb. pressure for flooding the eliminator surfaces.

The *American Blower Company* uses about 40 sq. ft. of scrubber surface per 1000 cu. ft. of air per minute, spaced 1 in. with 35-degree deflection of the air. These are made with plates having three corrugations and a final stamped lip. The eliminator surfaces are in three rows, the final row spaced $3\frac{1}{4}$ in., and designed on the basis of 12 sq. ft. of surface per 100 cu. ft. of air per minute. The spray nozzles are designed to give a uniformly distributed spray across the spray chamber, and the spray is in the direction of the air flow.

Using the Carrier design, the eliminator surface, spaced on $1\frac{1}{8}$-in. centers, becomes:

$$(132 \div 1\tfrac{1}{8}) \times 6 \times 2.63 \times 4 \div 12 = 617 \text{ sq. ft.}$$

The required amount is 427 sq. ft., or there is about 44 per cent excess area over the requirements.

The Baudalot Cooler.—The figure shows the design with the water cooler directly below the spray cooler chamber. In cooling water great care must be exercised to safeguard the pipes against freezing, and so the Baudalot cooler is used instead of the double-pipe cooler or the shell and tube type which would be preferred usually when cooling a non-freezing liquid. Such Baudalot coolers are usually designed on the basis of 35 lin. ft. of $1\frac{1}{4}$-in. pipe per ton of refrigeration. The problem requires a refrigeration of 10,560 B.t.u. per minute, or 52.8 tons. The surface, then, is

$$35 \times 52.8 = 1850 \text{ ft. of } 1\tfrac{1}{4}\text{-in. pipe.}$$

It is doubtful whether a cooler more than 8 ft. long could be arranged for with satisfaction in this type of design, so using 14 pipes high and pipes 8 ft. long the number of stands becomes $1850 \div (8 \times 14) = 16.5$, and this can be arranged for easily on 6- or 7-in. centers. Figure 328 gives a good idea of the entire spray design.

The problem has been worked out for the simplest design, i.e., a self-contained Baudalot cooler for the water, using direct expansion in these pipes for the ammonia; the entire spray chamber being located inside the room to be cooled. Without question the spray chamber could be placed in any convenient location and the Baudalot cooler could have been cooled with the use of carbon dioxide or brine could have been pumped in from an adjacent building.

Cold Air Ventilating Ducts.—In general, for processes using the spray chamber, such as gelatine or other industrial work, it is desired to keep a product exposed to

cool air and yet to air dry enough to prevent the surface of the product from having moisture condense on it. This means the selection of the proper operating temperature, relative humidity and corresponding dew point temperature, as has been outlined already. It is necessary to supply the chilled air as uniformly as possible in order that the room temperature will be uniform throughout. This is best done by some duct system, as indicated by Fig. 327. The problem requires the circulating of 22,000 cu. ft. of air per minute at a dew point temperature of 50.7 deg. F. The problem consists of finding the duct diameters at the different sections and the total head required of the fan to give the desired circulation.

In order to calculate the friction head due to the flow of air in ducts it is convenient to use a chart as is shown in Fig. 330, showing the resistance to the flow of air at a standard temperature and pressure at different velocities of flow in the pipe. The general formula for the loss of head due to the flow of a gas in pipes and ducts is:

$$h_a = f\frac{LP}{A} \times \frac{V^2}{2g},$$

where

h_a = loss of head due to flow in feet of the medium flowing (air in this case);
h = loss of head in inches of water;
f = coefficient of friction;
L = length of pipe or duct, in feet;
P = perimeter of pipe or duct, in feet;
A = cross-sectional area of pipe or duct, in feet.
$\frac{V^2}{2g}$ = velocity head;
V = average velocity, in feet per second;
d_a = density of air (0.075), lb. per cubic foot;
d_w = density of water (62.4 lb.), per cubic foot;
g = acceleration due to gravity.

As it is customary to state the head in inches of water, the formula may be expressed as:

$$h = h_a \times \frac{12 \times d_a}{d_w} = f\frac{12d_aLPV^2}{d_wA2g},$$

which is simply an expression for the ratio of the densities of the two substances. Therefore the second expression gives the loss of head in inches of water. Using an average value for f of 0.0036 for air velocities of from 1000 to 3000 ft. per minute, and with L = 100 ft., the formula becomes

$$h = \frac{12 \times 0.075 \times 100 \times 0.0036}{62.4 \times 64.32} \times \frac{P}{A}V^2,$$

and for round ducts $P/A = 4/D$, and therefore

$$h = 0.000323\frac{V^2}{D}$$

(where D is the diameter of the duct, in feet).

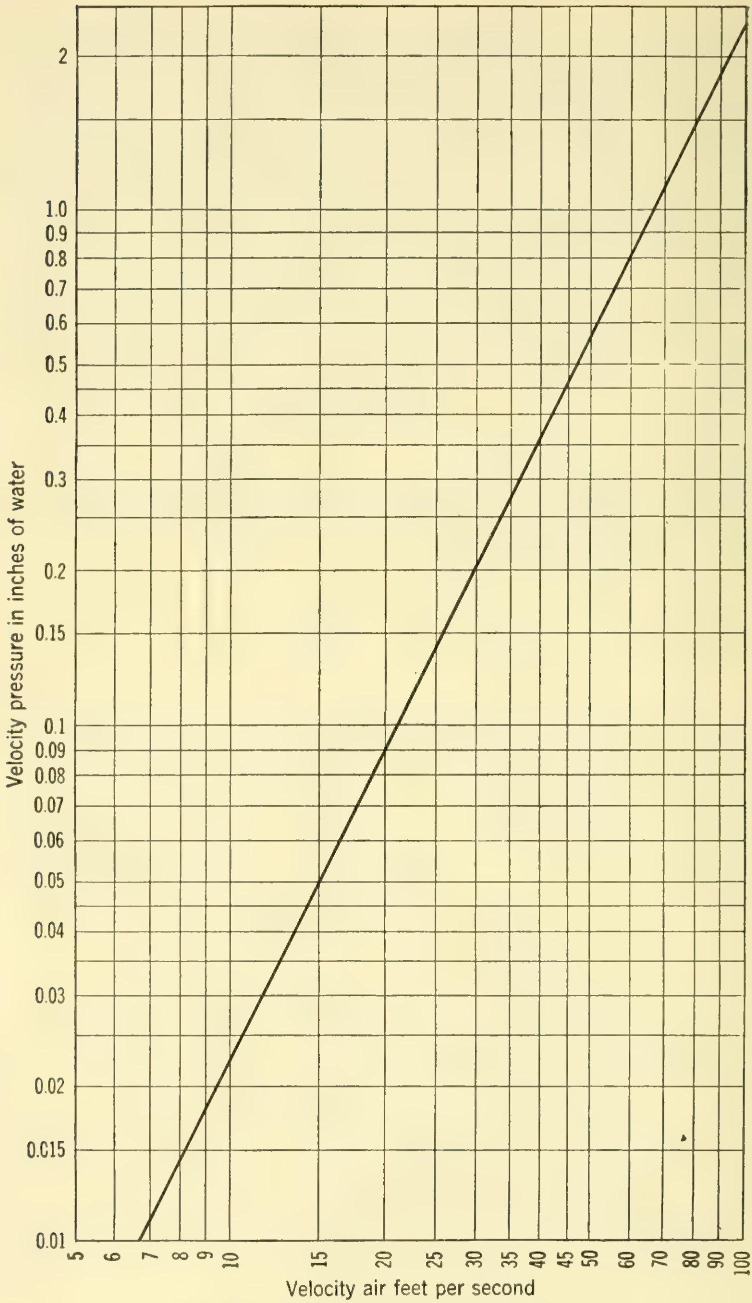


Fig. 329.—Velocity Pressure Head for Air.

If Q is the cubic feet per minute of air flowing, then $v = \frac{Q}{60A}$, and, making this substitution, the formula finally becomes

$$Q = 6678h^{\frac{1}{2}}\left(\frac{A^3}{P}\right)^{\frac{1}{2}},$$

and, if the duct is round, then

$$Q = 2625h^{\frac{1}{2}}d^{\frac{5}{2}} \quad \text{(for a duct 100 ft. long, and where } f = 0.0036\text{)}.$$

In the figure it will be noted that the quantity of air (in cubic feet per minute) is represented by the horizontal lines and that the resistance to the flow in inches of water per 100 ft of length is given by the vertical lines, whereas the different sizes of round ducts or the width of duct for rectangular cross-sections is given by the lines sloping upwards to the right.

In all work dealing with the flow of fluids, the terms total, static, and velocity are used. The static pressure is that which exerts a bursting pressure on the walls of the containing vessel, and is the amount necessary to overcome the resistance to the flow of the fluid. The velocity pressure is that pressure which would be produced by the velocity if the air were brought to rest without shock or impact and which has to be used to produce this velocity. The total pressure is the sum of the two, i.e., the static and the velocity pressures. The velocity pressure, coming from the velocity and the density of the air, is given for 70 deg. F. by the following and from Fig. 329.

TABLE 102

Velocity in Feet per Minute	Velocity Head
750	0.004
1200	0.09
1500	0.14
1800	0.21
2100	0.28
2400	0.37
2700	0.46
3000	0.57

In designing a duct system there are two general methods of procedure: to fix on the velocity of the air in the different parts of the system taken from standard practice, and to find the total pressure drop by adding up the resistances to the flow in the different sections or to select a suitable friction pressure drop near the outlet and keep this constant by selecting the duct sizes in the separate sections on the basis of this constant pressure drop and the required quantity of air flowing. The second method is the most convenient and the quickest, and its application is shown by the following:

Referring to Fig. 327 it will be seen that 18 outlets are to be used. The amount of air per outlet is 22,000 ÷ 18 = 1220 cu. ft. per minute and the velocity through a 14-in. outlet is 1135 ft. per minute. Charts for air flow are usually made out for 100 ft. of straight pipe or duct and for ideally perfect construction and freedom from flaws. To allow for rough edges, or other faulty construction, a factor is usually added, frequently 50 per cent. In addition, changes in the cross-section or the direction of flow increase the resistance and this increase is usually given in the percentage of the velocity head. Variations such as are usual are given in Fig. 331. Using the chart and the figure of the duct layout, Table 103 is obtained.

TABLE 103

Section	Volume of Air in Section, Cubic Feet per Minute	Diameter of Duct in Inches	Velocity of Air in Feet per Minute	Length of Section and Allowance for Ells, Etc.
a	2,440	17	1550	22 ft. + 0.09 × 0.17 in. of water
b	4,860	$22\frac{1}{4}$	1790	22 ft.
c	7,290	$26\frac{1}{4}$	1920	22 ft.
d	9,720	30	1980	22 ft. + 0.15 × 0.23 in. of water.
e	22,000	42	2390	44 ft.
Total	....	....	....	132 ft. + 0.049 in. of water.

Taking a uniform resistance to air flow of 0.14 in. of water per 100 ft. of duct, the total resistance will be (132 ÷ 100 × 0.14) + 0.049 = 0.234 in. of water, and with an allowance for roughness, poor workmanship, etc., of 50 per cent. the resistance will be 0.234 × 1.5 = 0.351 in. of water for the duct only. In addition, there will be the resistance to the flow in the spray chamber of 0.25 in., and there will be the residual velocity of the air leaving the duct which will require, for 1140 ft. per minute velocity of the air in the branch, a static pressure of 0.090 in. of water, making a total pressure on the fan of 0.691. In designing a duct system it is usual to provide for the return as well as the delivery of the air. Quite frequently the return duct will be practically a duplicate of the delivery duct, in which there will be an additional friction to overcome and a correspondingly greater pressure. In the present problem, as the entire plant is located in the same room, it does not appear to be necessary to put in the return duct. The design shown will work satisfactorily.

In the present problem the temperature of the air is 51 deg. F. in the duct, and the calculations for the total pressure must be increased by an amount sufficient to allow for the increased density over that at the standard temperature of 70 deg. F., which is 1.021 in this case. Using this multiplying factor the total pressure head becomes 0.691 × 1.021 = 0.706 in. of water. The static pressure at the fan is some percentage of the total pressure, and taking the ratio of the total to the static pressure as 1.26 the static pressure becomes 0.706 ÷ 1.26 = 0.560. A fan selected for this static pressure will be chosen as well as a capacity of 22,000 cu. ft.

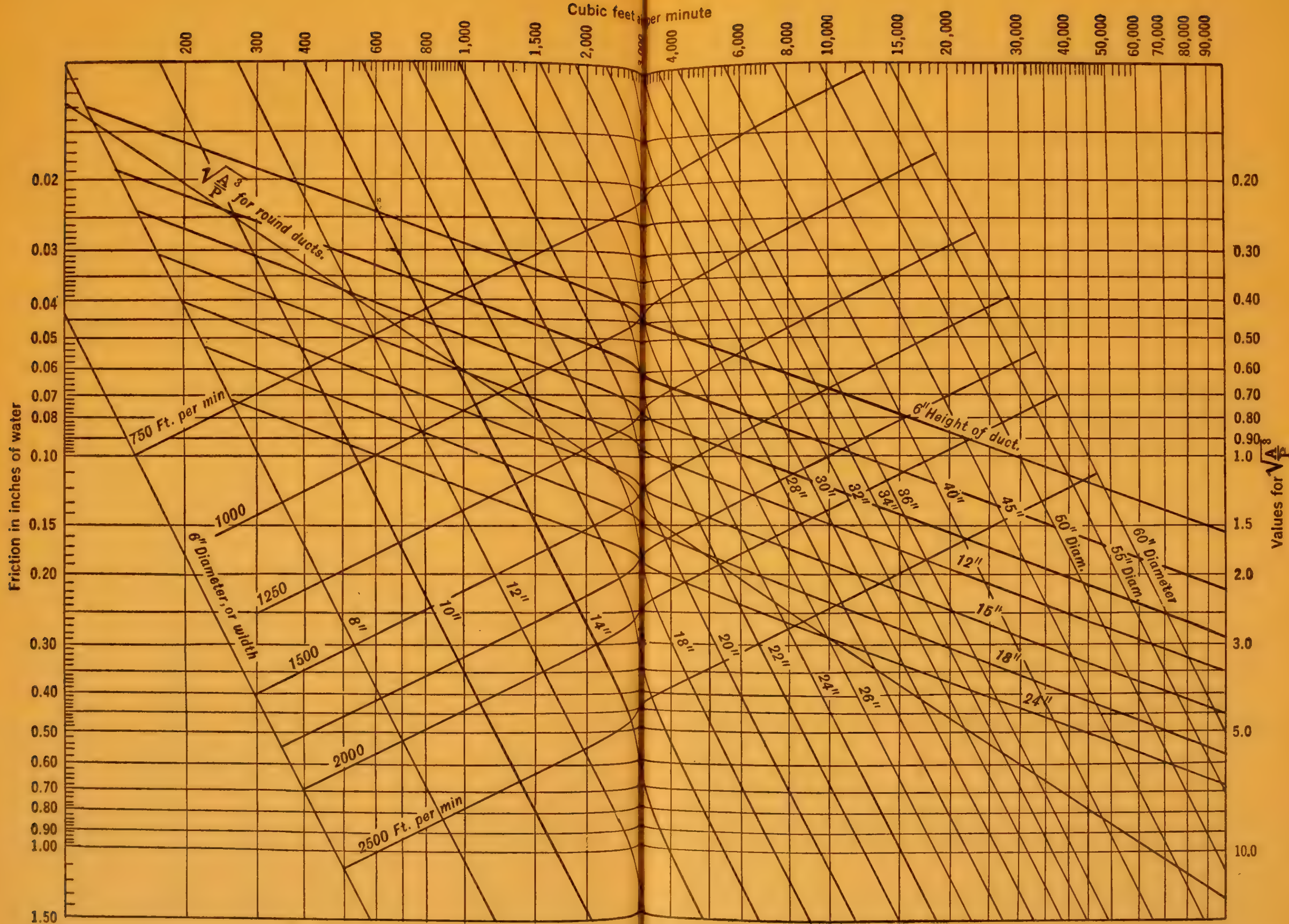


Fig. 330.—Drop of Pressure Due [to F]low of Air in Ducts in Inches of Water.

To face page 521.

per minute. It is well to note that the tabulated values for fan performance are given at 70 deg. F. For operation at other temperatures the following table will be of value:

TABLE 104

Temperature of the Air	Dividing Factor	Multiplying Factor
60	0.99	1.010
50	0.98	1.021
40	0.97	1.032
30	0.96	1.042
20	0.95	1.053

It will be noticed that the velocities shown in the calculation increased as the size of the duct increased; from a little more than 1600 ft. at 17 in. diameter to a little less than 2400 ft. per minute at 42 in. diameter of duct. Also that the 30-degree branch outlet resulted in the same loss of head as would have been experienced with nearly 24 ft. of 17 in. diameter duct. Too much emphasis cannot be given to the desirability of making the number of changes in the cross-section of the duct a minimum. Right-angled turns should be avoided at all times, and when the size of the duct is to be changed it should be done gradually.

In the problem a round duct was chosen. Should it be desired to use a rectangular one with the same resistance per foot of length the same chart as for the round ducts can be used, making use of the relation $Q = 6678h^{\frac{1}{2}}\left(\frac{A^3}{P}\right)^{\frac{1}{2}}$. For example, Fig. 330 has a line labeled $\left(\frac{A^3}{P}\right)^{\frac{1}{2}}$. To illustrate the use of the chart, suppose it is required to find the height of a rectangular duct if the width of the b section is taken as 30 in. The quantity of air flowing in this section is 4880 cu. ft. per minute. Take the same resistance as before, 0.14 in. per 100 ft., and continue upwards on the chart until the quantity 4880 is reached. Then follow parallel with the diameter lines until the line $\left(\frac{A^3}{P}\right)^{\frac{1}{2}}$ is reached and then vertically upwards until the width of duct of 30 in. is reached. This point gives the desired answer of 16 in. for the height of the duct.

Air Cooling in Blast Furnaces.—Some three or four installations of the "Gayley Dry Air Blast" process were made in the United States and one in Wales previous to 1910. The Gayley process was an attempt to increase the economy of the furnace and was successful in certain respects, but the process has not been continued because other economies in other directions were found to be as good but not so expensive in first and operating costs. However, as an example of a method to be applied to refrigeration the following will be of interest.

The Gayley dry-blast process was developed about 1900. It consisted in cooling the air to a dew point temperature of 25 deg. F. at atmospheric pressure at which time the moisture content would be 19.5 grains per pound of dry air. The claims made for the process are that the furnace can work with greater regularity, and that a reduction of $12\frac{1}{2}$ per cent of the fuel required and a gain in 10 per cent in the production of the furnace is possible by its use. The reason advanced for this phenomenal gain in economy is that the moisture in the air (which at times may be 150 grains per pound of dry air) chills off the fire, just when the furnace temperature should be the hottest, by the chemical action of breaking up the water vapor into oxygen and hydrogen. The result

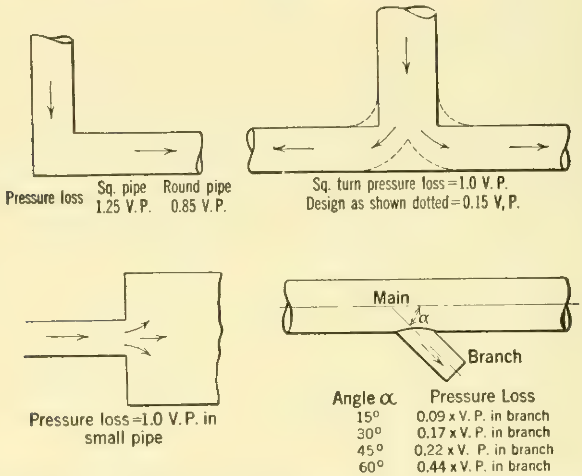


FIG. 331.—Loss of Velocity Head Due to Change of Section or Direction of Flow.

of this endothermic action is that there is not the proper removal of the sulphur or the proper deoxidation of the silicon during warm damp weather unless there is additional heat provided in the furnace by the addition of extra coke.

In the blast furnace it is necessary to use large blowing engines in order to compress the air required for supporting the combustion action in the furnace, and the blast may be at some pressure between 15 and 30 lb. gage. The compressed air is made to pass through checker work, previously heated by the exhaust gases from the furnace, until the blast temperature has become nearly 3000 deg. F. The refrigeration has been supplied as a rule on the suction side of the blowing engine, and therefore under these circumstances it will take place at approximately atmospheric pressure. The air cooled to 25 deg. F., with a uniform

water content of 19.5 grains per pound of dry air, is then passed to the compressor which is reduced in size thereby, on account of the increased density of the air.

The manner of cooling the air at the Isabella Works of the Carnegie Steel Company, but not operated since 1912, was to have two large bunker rooms with ammonia direct expansion piping in each, but so arranged as to permit the air to pass through one and then the other of these rooms by the operation of large butterfly valves, the reversal of the direction of flow thereby permitting the frost accumulation on the pipes to melt off in a manner similar to the present-time dehydrator of high pressure air for can ice agitation. Since 1912 the design of air cooling has changed decidedly so as to use the principle of the cooling tower or scrubber, arranged with tile wetted with water in the first stage and by brine in the second stage. Later came the design using sprays. This latter process, used in the Northern Steel Company, is shown in Fig. 332. In this case the air is compressed by the blowing tubs first, and in consequence of the higher pressure the air does not have to be cooled as low in temperature to secure the same weight of water content. For example, by compressing the air to 20 lb. gage it has been found necessary to cool the air down only to 46 deg. F. in order to secure the same weight of moisture as would be secured by cooling the air at atmospheric pressure down to 25 deg. F. In this case water can be used in the spray nozzles for the entire cooling action, and there will not be any necessity for reconcentration as would be required in the case of brine. Refrigeration is more economical because of the higher suction pressure being carried.

The principle underlying the liquefaction under pressure is as follows: Take a condition of air, saturated with water vapor, at 41 deg. F., at one atmosphere.

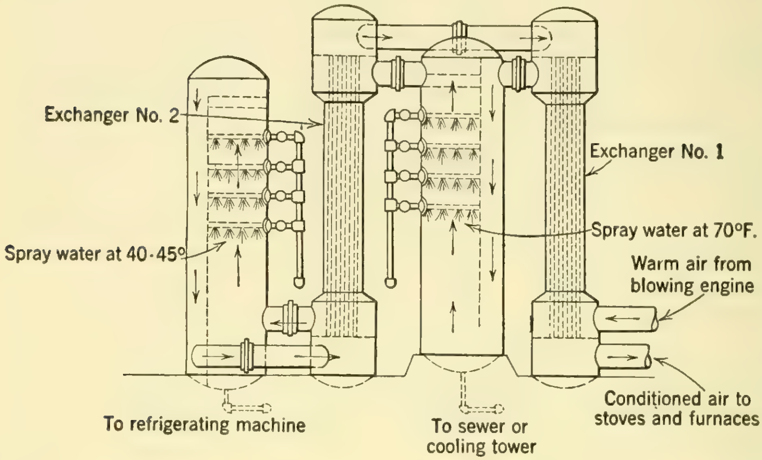
Pressure of water vapor (at 41 deg. F.)	= 0.127 lb.
Pressure of the air = 14.69 − 0.127	= 14.56 lb.
Grains of moisture per 1 lb. of dry air	= 37.87
Volume of 1 lb. dry air	= 12.62 cu. ft.

Now let the temperature remain constant at 41 deg. F. and compress the mixture up to 10 lb. gage or 24.69 lb. absolute, the air remaining saturated with moisture all the time. Then 24.69 − 0.13 = 24.56 lb. is the new pressure of the air content of the mixture.

$$\text{The new air volume} = 12.62 \times \frac{14.56}{24.56} = 7.49 \text{ cu. ft.}$$

The condensation of the water vapor occurs in the same ratio or

$$37.87 \times \frac{14.56}{24.56} = 22.45 \text{ grains.}$$



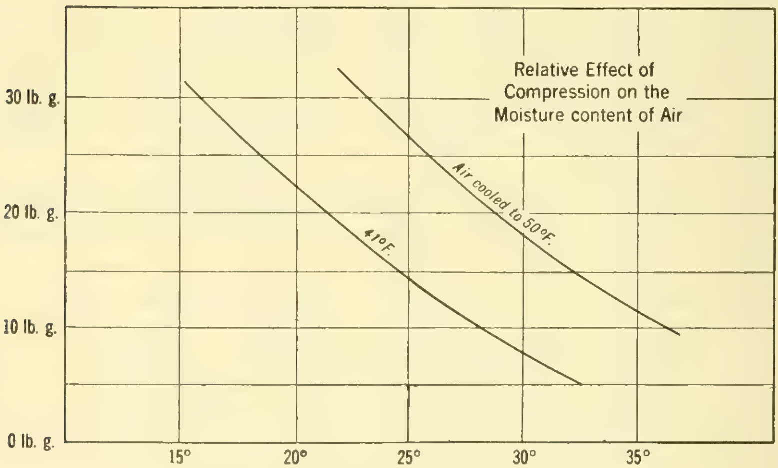


FIG. 332.—Air Cooling under Pressure.

However, this moisture content, 22.45 grains, is the amount which would give saturation with a pressure of one atmosphere air pressure and a temperature of a little more than 28 deg. F. Figure 332 gives an idea of the effect of compression and moderate cooling on the water vapor in the air. The principle just outlined occurs also in the de-

humidifier in the manufacture of ice with high-pressure air agitation, and in the synthetic process for the manufacture of ammonia.

The Cooling of Theatres and Public Buildings.—Mechanical refrigeration has been able to provide a means of cooling the air so that the public dining room, banquet hall and theatre are reasonably comfortable and suitable for service during the hottest part of the summer. The same may be said also in regard to banks, offices and such places of congested gathering as the New York Stock Exchange, etc. In none of these places is there a desire to chill the air very much, but simply to make it feel cool on entering. Therefore some such plan as the following would be used: The temperature of the chilled air will be 80 degrees when the outside air is 90, 75 when the outside air is 85, and 72 when the outside air temperature is 80 deg. F., and the relative humidity will not be greater than 60 per cent when the temperature in the theatre is 80 degrees and 65 per cent when the air is 72 degrees. In none of these cases is the air cooled more than 10 deg. F., and the result is a feeling of comfort on entering the building, but there must never be a sensation of chill. The procedure in such cases of cooling is to use the same equipment for ventilation and cooling as for heating and ventilation, the air being washed and a certain percentage, 50 to 67 per cent, being circulated. Such cooling plants are of large capacities, requiring as they do about $2\frac{1}{2}$ tons of refrigeration per 1000 cu. ft. per minute of fan capacity for localities similar to the summer conditions found in Chicago.

The Refrigerating Machine.—The refrigerating machine is now (1928) always a compressor, and the refrigerant has nearly always been carbon dioxide because of the lack of danger due to its use.[4] As the air in the spray chamber is seldom, if ever, cooled below 55 deg. F. (using 47-degree water in the sprays), the refrigerating machine can operate at the high boiling temperature of 30 to 32 deg. F., and so in consequence of which the capacity of the compressor is high as compared with the standard conditions of 5 deg. F. and the horse power per ton of refrigeration is reduced. For some time the method of cooling has been to install direct expansion piping at the entrance of the spray chamber using an amount equal to 35 lin. ft. of $1\frac{1}{4}$-in. pipe per ton of refrigeration. The piping is used both to cool the entering air and also to cool the water which is showered over the coils and which act as a " Baudalot " cooler. With such an arrangement it would be out of the question to use ammonia because of the possible panic should some accident occur to these coils and ammonia gas enter the ventilating ducts.

[4] Tests reported to have been performed by the Board of Trade on the SS. *Braemar Castle* indicated that no discomfort was experienced when the entire charge of carbon dioxide was released in the engine room, nor were lighted candles affected in any manner.

The air leaving the spray chamber will have an assumed temperature of 55 degrees and 100 per cent relative humidity, but during the passage through the fan and the distributing ducts the temperature will rise about 8 degrees, depending on the friction losses and heat leakage,

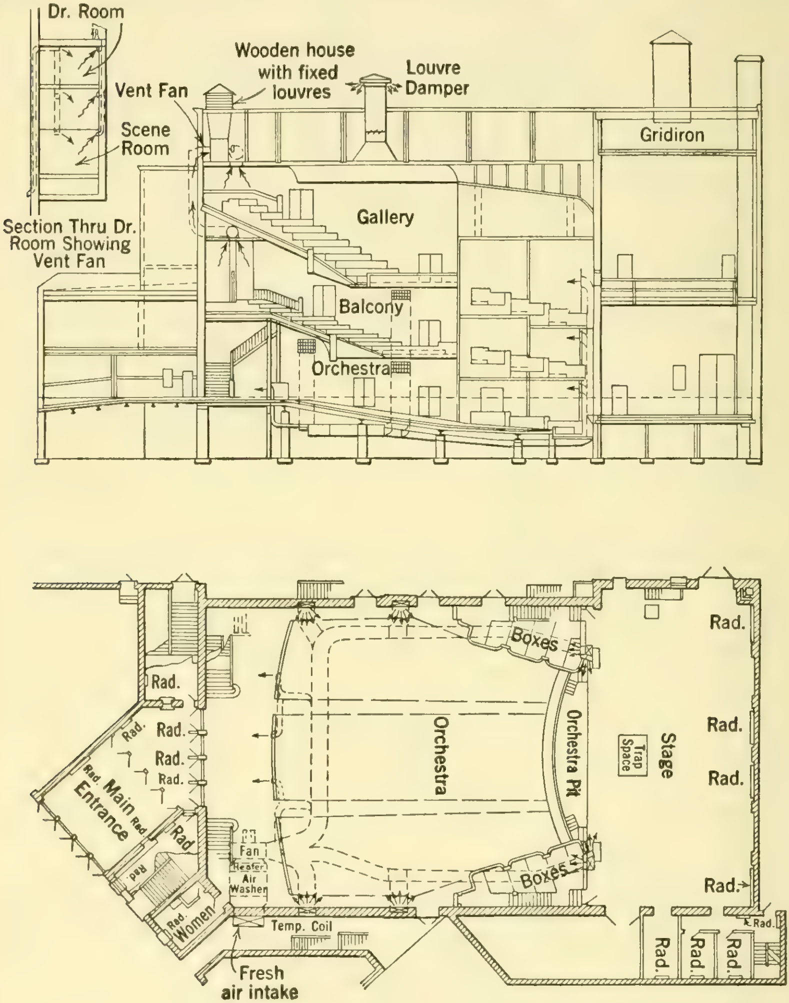


FIG. 333.—Auditorium Cooling and Ventilation.

which will give the air a dry bulb temperature of 63 degrees and a relative humidity of 75 per cent at the point of entrance to the hall or the auditorium. The air may be admitted to the auditorium either by the use of " mushrooms " in the floor or by the use of ceiling registers

and downward draft and in either case about 35 cu. ft. of air per minute per person is supplied. The fresh air from the sprays mixes with the air in the auditorium, and, if the conditions are such as the installation was designed for, the temperature will be 80 degrees and 60 per cent relative humidity. In the case of banquet halls and special dining rooms, the air can be designed to enter through registers in the walls, some 6 or 7 ft. from the floor, or from registers in the columns.

The Spray Chamber.—The air is cooled always by the use of water properly atomized by passing through nozzles especially designed for the purpose, usually with 15 to 25 lb. static pressure on the nozzle. Some engineers use the rule 8 gal. of water per 1000 cu. ft. of air per minute, while others use the rule of 1 lb. of water per 1 lb. of air. If the water is permitted to rise 8 deg. F. in temperature, 25 lb. or 8 gal. per minute would be required. A "Carrier" nozzle with a $\frac{3}{16}$-in. opening will deliver 15 lb. of water at a pressure of 20 lb. per sq. in., and 17 lb. at a pressure of 25 lb. per sq. in., whereas a $\frac{1}{4}$-in. opening will deliver 26 and 29 lb., respectively, at these pressures. From Table 101 it will be be seen that the Sturtevant and the Webster nozzles have about the same capacities. The details of the spray chamber are the same as in the previous problem.

Problem.—It is desired to install a cooling plant in a 3000-seat theatre for summer conditions of 90 deg. F. dry bulb and 60 per cent relative humidity outside conditions, 80 degree dry bulb inside and 63 degree dew point temperature. The heating effect of the illumination will be taken as 1500 B.t.u. per minute, and the heat leakage through the roof, walls, etc., will be taken as 3000 B.t.u. per minute. The sensible heat will be taken at 7.0 B.t.u. per minute per person, and the amount of water vapor emitted as 18.9 grains. This last is a high value but it is justified during hot weather. These heating values may be summed up as follows:

	B.t.u.
a. The heating effect of the audience (the sensible heat) = 7.0 × 3000	= 21,000
b. The heat leakage (assumed)	= 3,000
c. The illumination (assumed)	= 1,500
d. The heat equivalent of the work of the fan (estimated at 45 hp.)	= 1,910
Total refrigeration per minute	= 27,410

If the specific heat of air at 63.0 degrees, saturated with moisture, is taken as 0.2472, and the rise of temperature of the air is 17.0 deg. F., the heat capacity of the air will be 17.0 × 0.2472 = 4.20 B.t.u. The total amount of air to be circulated by the fan to absorb 27,410 B.t.u. per minute is therefore 6526 lb. The volume of 1 lb. of air saturated with moisture at 63.0 degrees is 13.43 cu. ft., so that the total capacity of the fan per minute is 6526 × 13.43 or 87,640 cu. ft., and the amount supplied to each person, if at 63.0 degrees, is 29.2 cu. ft. per minute.

The Refrigeration Required.—The amount of water vapor emitted is 18.9 × 3000 ÷ 7000 = 8.1 lb. per minute, and the refrigeration required to condense this amount will be 8.1 × 1055 = 8546 B.t.u., thereby making a total of 35,956 B.t.u.

per minute, which would have to be removed from the air if all of it were to be recirculated. As one half is to be recirculated and the other half is exhausted into the atmosphere there will be 17,980 B.t.u. per minute of refrigeration required for

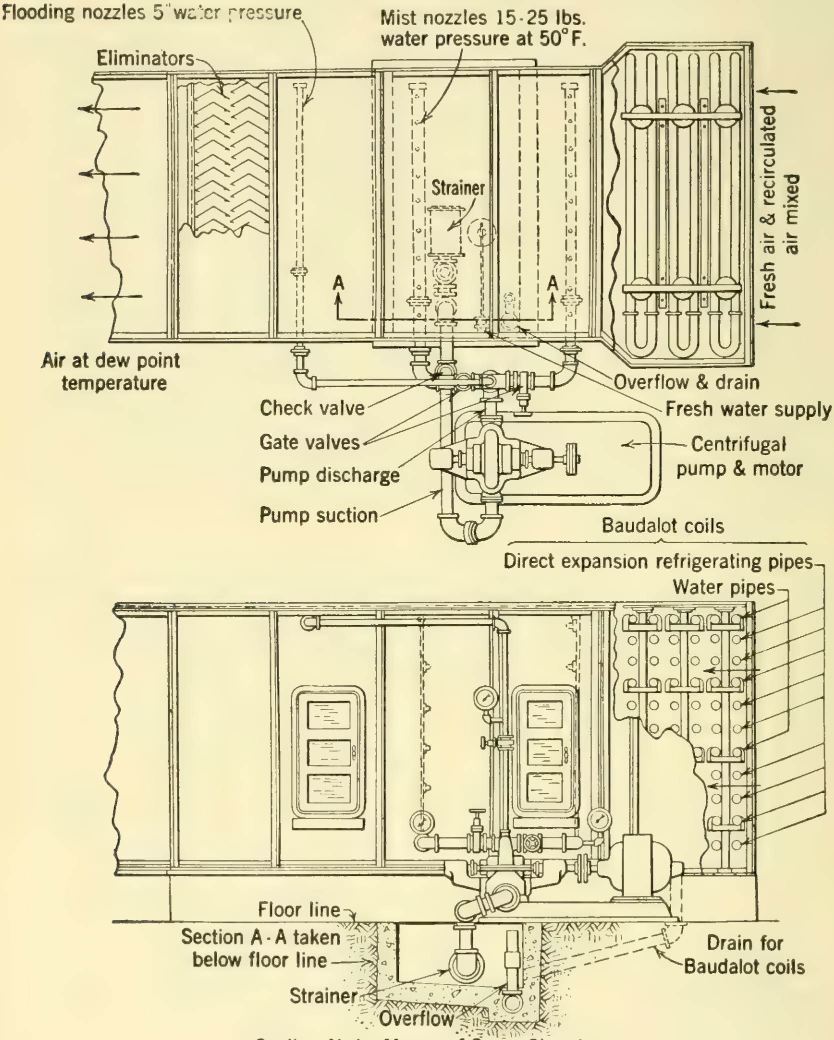


Fig. 334.—The Water Spray Air Cooling Chamber with Direct Expansion Coils for Carbon Dioxide Refrigeration.

the air to be recirculated. The fresh air will enter the spray chamber at 90 degrees and 60 per cent relative humidity, which corresponds to a dew point temperature of 74.2 and a moisture content of 127 grains per pound of dry air. The refrigeration

required to cool this air to the temperature of the spray chamber, taken from the total heat curve is: $40.9 - 28.0 = 12.9$ B.t.u. per pound, the total refrigeration is $6526 \div 2 \times 12.9 = 42{,}090$ B.t.u., and the total demand on the refrigeration equipment becomes $42{,}090 + 17{,}980 = 60{,}070$ B.t.u. Adding 5 per cent for safety, the amount becomes 63,070 B.t.u. per minute, or 315 tons of refrigeration.

It is clear from the calculation that the recirculated air, for the conditions stated in the problem, requires less cooling effect than does the fresh air from the outside, and that the amount of recirculation affects the load on the machine very decidedly. Therefore the operator has some reserve capacity in the machine available, on a pinch, by recirculating more air, if permitted by the city ordinances, as, for example, the circulation of 67 per cent instead of only 50 per cent.

If the spray water has a rise of temperature of 8 degrees there will be required 3 gal. per ton of refrigeration per minute, or $3.0 \times 315 = 945$ gal. total per

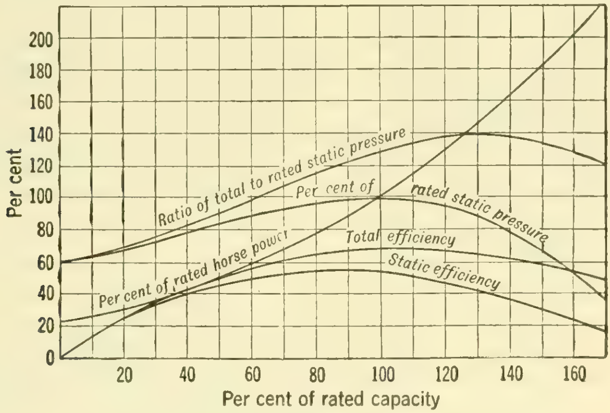


FIG. 335.—Characteristics of Conoidal Fans.

minute. Usually there is provided an additional amount, say 20 per cent, so that the pump will need to be 1134 gal. If a ¼-in. nozzle is used, which has a capacity of 29 lb. of water per minute with a pressure of 25 lb. per sq. in. at the nozzle, the number of sprays required will be $\frac{1134 \times 8.33}{29} = 326$. Using a velocity of 500 ft. per minute for the air in the spray chamber there will be required $87{,}640 \div 500 = 175.2$ sq. ft. of cross-sectional area, and if the spray chamber is limited to 10 ft. in height the width would need to be 17.5 ft. The arrangement of the nozzles, the amount of evaporating surface for cooling the water, the amount of eliminator surface, etc., will be found in a manner similar to that in the previous problem.

Air Conditioning in Mines.—According to Wilcox and Farmer,[5] the only mechanically refrigerated mine in the world is located in the Province of Minas Geraes, in Brazil. This mine has its pit head 2770 ft. above sea level and the mine depth in 1920 had been carried to 6400 ft.

[5] F. A. Wilcox and J. D. Farmer, Application of Refrigeration to the Ventilation of Mines, Fourth International Congress of Refrigeration.

where the rock temperature was recorded to be 118 deg. F. and the wet bulb temperature was from 90 to 95 deg. F., thereby making operating conditions very difficult for the workmen. Calculations showed that at a mine depth of 6500 ft., in order to insure not more than 81 degrees wet bulb temperature, the initial dew point temperature of the air would need to be 45.5 degrees, which would correspond to

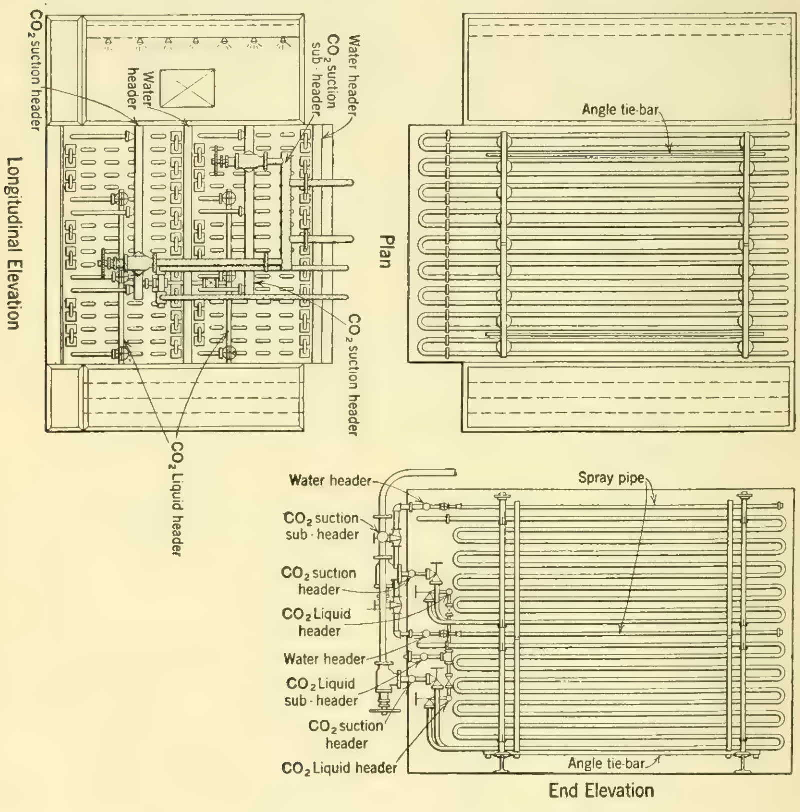


FIG. 336.—Air Cooling with Carbon Dioxide.

50 grains of water per one pound of dry air at 27.4 in. of the barometer. These calculations were based on a 34-in. barometer, a wet bulb temperature of 81 degrees, and a moisture content of 110 grains.

The plant selected was based on a fan capacity of 80,000 cu. ft. per minute and the air to be cooled to a dew point temperature of 43 deg. F., which represented a removal of 100,600 B.t.u. per minute, or 500 tons of refrigeration capacity. The load was arranged for with six units in order to permit economical operation at all seasons of the year, especially

at those times when the load would be less than the maximum. In this case water can be used satisfactorily for the cooling of the air, with a water temperature range of from 3 to 5 degrees. The method of cooling used in this installation was by the use of revolving cylinders whereby the air was brought into contact with a film of chilled water. The rock temperature, according to the mine records, was cooled from 118 to 107 deg. F.

Carburetor Testing.—Some manufacturers of automobiles have a cold temperature room in which a motor is operated under varying loads with various designs and adjustments of carburetors.[6] The cold temperature room is designed for − 10 deg. F. The automobile motor usually has the dynamometer outside the room and the cooling water for the radiator sometimes is piped in from the outside. The refrigeration required to maintain the temperature constant at − 10 degrees for a 60 hp. motor, may be calculated as follows, assuming an electrical dynamometer, 90 degrees outside, 6 in. of corkboard on the walls, ceiling and floors of a room 12 by 12 by 7 ft. Assume the walls are of 6-in. concrete. The heat balance of the motor may be taken as

	Per cent
Engine friction	4
Radiation	4
Exhaust	40
Jacket	27
Useful work	25

Even with the exhaust piping well lagged some heat will be lost to the room, and this will be taken as 25 per cent. Taking the efficiency of the electric dynamometer as 95 per cent, 5 per cent of the brake hp. will appear as heat in the electric windings. The amount of air required, to be brought in from the outside, will be taken as 15 lb. per pound of gasoline, and 0.7 pint of gasoline[7] will be required per b.hp. hour. The heat entering the room is then

$$Q = (0.04 + 0.04 + 0.25 \times 0.40)\left(\frac{60 \times 42.4}{0.25}\right) + (0.05 \times 60 \times 42.4)$$
$$+ \left(\frac{47.5 \times 30.7 \times 15}{60}\right) + \left(\frac{624 \times 0.05 \times 90 + 10}{60}\right)$$
$$= 2380 \text{ B.t.u. per minute} = 11.9 \text{ tons.}$$

[6] This problem was suggested by R. N. Cole, of Canton, Ohio.

[7] Take the specific gravity of gasoline as 0.702 and 1.0 pint of gasoline as weighing 0.732 lb.

Adding 10 per cent for safety, the total amount of refrigeration which should be provided will be 13.1 tons.

Selection of Fan.—The essential factors in the selection of a fan are: the capacity of the fan in cubic feet per minute; the static pressure developed at the fan discharge; and the horse power required to drive the fan. Each type of design has its own characteristic curves, and these are dependent on the curvature of the blade tip which may be radial, forward, backward and steel plate. The short curved blade is the

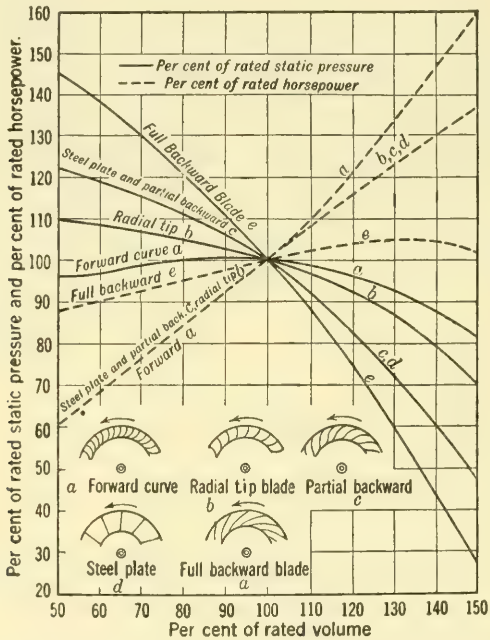


FIG. 337.—Characteristic Curves for Standard Fan Tips.

favorite with refrigerating engineers because of its ability to run at moderately high speeds.

The *steel plate* is for slow speeds only, and usually is steam engine driven. It is not an efficient type and is not used in refrigeration to any extent.

The *full forward* tip fan has a high outlet velocity but is noisy. It has the slowest tip speed for a given tip velocity of any design, but the power required increases rapidly with the speed, and it requires an oversized motor to prevent danger in the burning out of the motor windings.

The *full backward* tip fan has a rising horse power up to a certain

capacity and then drops off. The efficiency is high and this design is satisfactory in every respect for motor drives.

After the type of design of the fan is selected the size and the speed must be chosen. Previous calculations will have determined the total pressure, and the fan catalogue will show the ratio of the static to the total pressure, the static pressure can then be calculated and the problem reduced to the selection of a fan for a known capacity and static pressure at discharge. Reference to the tables will seldom give the exact conditions required, but other conditions may be found from the following:

a. The blade tip velocity and the volume of air delivered varies *directly* as the revolutions per minute.

b. The static pressure developed varies as the *square* of the revolutions per minute.

c. The power required to drive the fan varies as the *cube* of the revolutions per minute.

It is quite evident that increasing the speed in order to get the required volume of air should be limited to small amounts unless the increased static pressure is also desired. For example, the Niagara No. 6 conoidal fan at 472 r.p.m. delivers a volume of 13,450 cu. ft. per minute and requires 5.8 hp., but No. 8 at 204 r.p.m. delivers 13,820 cu. ft. and requires only 1.98 hp.

TABLE 105

CAPACITIES OF BUFFALO NIAGARA CONOIDAL FANS (TYPE N) UNDER AVERAGE WORKING CONDITIONS

70 Deg. F. and 29.92-In. Barometer

Size	Diameter of Blast Wheel, Inches	Area of Outlet, Square Feet	½-In. Static Pressure =0.288 Oz.			¾-In. Static Pressure =0.433 Oz.			1-In. Static Pressure =0.577 Oz.			1½-In. Static Pressure =0.865 Oz.		
			R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power
3	15⅝	1.31	544	1,945	0.28	668	2,380	0.51	770	2,750	0.78	943	3,365	1.45
3½	18⅛	1.79	465	2,642	0.38	572	3,240	0.69	660	3,740	1.06	809	4,580	1.97
4	20½	2.33	408	3,459	0.50	500	4,230	0.90	577	4,895	1.39	709	5,980	2.58
4½	23½	2.95	362	4,375	0.63	445	5,350	1.14	514	6,195	1.75	630	7,575	3.26
5	26⅛	3.64	326	5,400	0.77	400	6,610	1.41	462	7,645	2.16	566	9,350	4.03
5½	28¾	4.41	296	6,540	0.94	364	8,000	1.71	420	9,250	2.62	515	11,320	4.87
6	31⅜	5.25	272	7,780	1.11	334	9,525	2.03	386	11,000	3.12	472	13,450	5.80
7	36½	7.14	233	10,590	1.52	286	12,950	2.77	330	14,980	4.24	405	18,330	7.90
8	42	9.33	204	13,820	1.98	250	16,910	3.61	289	19,550	5.54	354	23,950	10.30
9	47	11.81	181	17,500	2.51	222	21,400	4.57	256	24,750	7.01	314	30,300	13.05
10	52	14.58	103	21,600	3.09	200	26,450	5.65	231	30,550	8.65	283	37,400	16.10
11	58	17.64	148	26,150	3.74	182	32,000	6.85	210	37,000	10.48	257	45,250	19.48
12	63	21.00	136	31,100	4.45	167	38,100	8.15	193	44,050	12.48	236	53,900	23.20
13	68	24.65	125	36,500	5.22	154	44,700	9.56	178	51,650	14.62	217	63,200	27.20
14	73	28.68	116	42,350	6.06	143	51,900	11.08	165	60,000	16.96	202	73,200	31.55
15	78	32.80	109	48,550	6.95	133	59,500	12.70	154	68,850	19.49	189	84,100	36.25
16	84	37.32	102	55,300	7.91	125	67,750	14.46	144	78,300	22.15	177	95,750	41.20
17	89	42.14	96	62,500	8.95	118	76,500	16.32	136	88,400	25.00	167	108,000	46.50
18	94	42.24	91	70,000	10.01	111	85,600	18.30	128	99,100	28.05	157	121,200	52.15
19	99	52.63	86	78,000	11.15	105	95,500	20.40	122	110,200	31.25	149	135,000	58.05
20	105	58.32	82	86,450	12.36	100	105,850	22.60	116	122,200	34.65	142	149,500	64.45

TABLE 105—*Continued*

Size	Diameter of Blast Wheel, Inches	Area of Outlet, Square Feet	2-In. Static Pressure = 1.154 Oz.			2½-In. Static Pressure = 1.442 Oz.			3-In. Static Pressure = 1.734 Oz.			3½-In. Static Pressure = 2.019 Oz.		
			R.p.m.	Volume of cubic feet per minute	Horsepower	R.p.m.	Volume of cubic feet per minute	Horsepower	R.p.m.	Volume of cubic feet per minute	Horsepower	R.p.m.	Volume of cubic feet per minute	Horsepower
3	$15\frac{5}{8}$	1.31	1088	3,890	2.21	1215	4,350	3.08	1332	4,770	4.05	1443	5,150	5.13
$3\frac{1}{2}$	$18\frac{1}{8}$	1.79	934	5,300	3.01	1010	5,930	4.19	1141	6,495	5.53	1238	7,010	6.98
4	$20\frac{1}{2}$	2.33	817	6,920	3.93	912	7,730	5.47	1000	8,480	7.22	1082	9,160	9.12
$4\frac{1}{2}$	$23\frac{1}{2}$	2.95	726	8,750	4.97	810	9,795	6.93	890	10,740	9.14	964	11,590	11.55
5	$26\frac{1}{8}$	3.64	655	10,820	6.15	730	12,070	8.55	800	13,250	11.26	868	14,300	14.25
$5\frac{1}{2}$	$28\frac{3}{4}$	4.41	595	13,100	7.43	664	14,600	10.35	728	16,030	13.62	789	17,300	17.25
6	$31\frac{3}{8}$	5.25	545	15,550	8.85	609	17,390	12.30	667	19,090	16.22	723	20,600	20.55
7	$36\frac{1}{2}$	7.14	468	21,200	12.02	522	23,650	16.75	572	26,000	22.10	620	28,050	27.95
8	42	9.33	409	27,650	15.70	456	30,900	21.90	500	33,950	28.85	542	36,600	36.50
9	47	11.81	364	35,050	19.90	405	39,100	27.70	445	42,950	36.55	482	46,350	46.20
10	52	14.58	327	43,250	24.55	365	48,300	34.20	400	53,000	45.15	433	57,200	57.00
11	58	17.64	297	52,300	29.70	332	58,450	41.45	364	64,100	54.60	394	69,300	69.00
12	63	21.00	272	62,300	35.50	304	69,550	49.25	334	76,400	65.00	361	82,500	82.15
13	68	24.65	252	73,050	41.50	280	81,600	57.80	308	89,550	76.30	334	96,750	96.45
14	73	28.68	234	84,900	48.15	261	94,600	67.05	286	103,900	88.70	310	112,050	111.90
15	78	32.80	218	97,250	55.25	243	108,700	77.00	267	119,200	101.50	289	128,800	128.20
16	84	37.32	204	110,750	62.85	228	123,600	87.50	250	135,800	115.50	271	146,400	146.00
17	89	42.14	192	125,000	71.00	214	139,500	99.00	235	153,100	130.30	255	165,300	164.80
18	94	47.24	182	140,000	79.50	203	156,500	110.80	222	171,800	146.00	241	185,300	184.60
19	99	52.63	172	156,000	88.55	192	174,200	123.40	211	191,200	162.80	228	206,200	206.00
20	105	58.32	164	173,000	98.25	183	193,000	136.80	200	212,000	180.30	217	229,000	228.00

Total Pressure is 127.4 per cent of the Rated Static Pressure.

TABLE 105*a*

CAPACITIES OF BUFFALO TURBO CONOIDAL FANS (TYPE T) UNDER AVERAGE WORKING CONDITIONS

70 Deg. F. and 29.92-In. Barometer

Size	Diameter of Blast Wheel, Inches	Area of Outlet, Square Feet	½-In. Static Pressure =0.288 Oz.			¾-In. Static Pressure =0.433 Oz.			1-In. Static Pressure =0.577 Oz.			1½-In. Static Pressure =0.865 Oz.		
			R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power
2½	14¼	0.91	1115	1,230	0.20	1368	1,500	0.36	1580	1,740	0.56	1935	2,120	1.03
3	17¼	1.31	930	1,770	0.28	1140	2,160	0.52	1315	2,500	0.81	1610	3,060	1.48
3½	20	1.79	797	2,410	0.39	976	2,940	0.71	1130	3,410	1.10	1380	4,160	2.02
4	22¾	2.33	697	3,140	0.51	855	3,850	0.93	987	4,450	1.44	1208	5,440	2.64
4½	25¾	2.95	620	3,980	0.64	760	4,860	1.18	879	5,640	1.82	1075	6,890	3.34
5	28½	3.64	558	4,910	0.79	684	6,000	1.45	790	6,950	2.25	966	8,500	4.12
5½	31½	4.41	507	5,950	0.96	621	7,270	1.76	719	8,400	2.72	880	10,300	5.00
6	34¼	5.25	465	7,070	1.14	570	8,650	2.09	658	10,000	3.24	806	12,230	5.77
6½	36¾	6.16	430	8,300	1.33	526	10,200	2.46	608	11,750	3.80	743	14,350	6.96
7	39¾	7.14	398	9,630	1.55	488	11,780	2.85	565	13,610	5.40	690	16,650	8.09
7½	42½	8.19	372	11,050	1.78	456	13,500	3.27	526	15,610	5.05	645	19,100	9.27
8	45½	9.33	349	12,590	2.02	428	15,370	3.72	495	17,800	5.75	604	21,750	10.55
8½	48	10.53	328	14,200	2.28	402	17,380	4.21	465	20,100	6.50	569	24,600	11.90
9	51¼	11.81	310	15,900	2.56	380	19,450	4.71	440	22,500	7.29	536	27,500	13.35
10	56¾	14.58	279	19,650	3.16	342	24,050	5.82	395	27,800	9.00	483	34,000	16.50
11	62½	17.64	254	23,800	3.82	311	29,100	7.05	359	33,700	10.90	439	41,100	19.95
12	68	21.00	232	28,300	4.55	286	34,600	8.40	329	40,100	12.95	402	49,000	23.80
13	73½	24.65	214	33,200	5.34	263	40,600	9.85	304	47,000	15.20	372	57,500	27.90
14	79	28.68	198	38,500	6.20	244	47,100	11.40	282	54,500	17.62	345	66,700	32.35
15	84¾	32.80	186	44,200	7.11	228	54,050	13.08	264	62,600	20.20	322	76,500	37.15
16	90¼	37.32	174	50,300	8.09	214	61,500	14.90	247	71,200	23.00	302	87,100	42.25

TABLE 105*a*—*Continued*

Size	Diameter of Blast Wheel, Inches	Area of Outlet, Square Feet	2-In. Static Pressure = 1.154 Oz.			2½-In. Static Pressure = 1.442 Oz.			3-In. Static Pressure = 1.734 Oz.			3½-In. Static Pressure = 2.019 Oz.		
			R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power
2½	14¼	0.91	2225	2,455	1.59	2490	2,750	2.22	2740	3,010	2.94	2958	3,250	3.69
3	17¼	1.31	1860	3,540	2.29	2075	3,960	3.19	2282	4,330	4.23	2463	4,680	5.30
3½	20	1.79	1595	4,800	3.12	1780	5,390	4.35	1958	5,890	5.75	2115	6,360	7.22
4	22¾	2.33	1395	6,270	4.08	1559	7,050	5.68	1713	7,700	7.52	1850	8,320	9.45
4½	25¾	2.95	1240	7,950	5.16	1385	8,920	7.19	1522	9,740	9.52	1645	10,550	11.95
5	28½	3.64	1117	9,800	6.37	1249	11,000	8.87	1370	12,000	11.75	1480	13,000	14.75
5½	31½	4.41	1015	11,880	7.72	1133	13,300	10.75	1245	14,550	14.23	1345	15,750	17.85
6	34¼	5.25	932	14,120	9.18	1040	15,800	12.78	1141	17,300	16.92	1232	18,700	21.25
6½	36¾	6.16	860	16,600	10.76	960	18,600	15.00	1054	20,300	19.85	1139	22,950	24.90
7	39¾	7.14	799	19,250	12.50	891	21,550	17.40	978	23,550	23.05	1056	25,450	28.90
7½	42½	8.19	745	22,100	14.32	831	24,750	19.95	914	27,050	26.40	987	29,200	33.20
8	45½	9.33	700	25,100	16.30	780	28,150	22.70	856	30,800	30.10	925	33,300	37.75
8½	48	10.53	657	28,400	18.40	736	31,800	25.60	807	34,750	33.95	870	37,550	42.25
9	51¼	11.81	621	31,800	20.65	693	35,600	28.75	761	38,950	38.05	822	41,050	47.80
10	56¾	14.58	559	39,300	25.50	625	44,000	35.50	685	48,100	47.00	740	52,000	59.00
11	62½	17.64	507	47,450	30.85	567	53,250	42.95	623	58,150	56.90	673	62,900	71.45
12	68	21.00	465	56,500	36.75	520	63,500	51.10	570	69,250	67.70	616	74,950	85.00
13	73½	24.65	430	66,200	43.05	480	74,400	60.00	527	81,300	79.40	569	87,900	99.60
14	79	28.68	399	76,800	50.00	445	86,300	69.55	489	94,300	92.10	528	101,900	115.60
15	84¾	32.80	373	88,500	57.40	415	99,000	80.00	456	108,000	105.70	493	117,000	133.00
16	90¼	37.32	349	100,500	65.30	390	112,500	90.90	428	123,000	120.50	462	133,000	151.00

Total Pressure is 122.7 per cent of the Rated Static Pressure.

TABLE 106

CAPACITIES OF BUFFALO PLANOIDAL STEEL PLATE BLOWERS (TYPE L) UNDER AVERAGE WORKING CONDITIONS

70 Deg. F. and 29.92-In. Barometer

Size	Diameter of Blast Wheel, Inches	Area of Outlet, Square Feet	½-In. Static Pressure =0.288 Oz.			¾-In. Static Pressure =0.433 Oz.			1-In. Static Pressure =0.577 Oz.			1½-In. Static Pressure =0.865 Oz.		
			R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power
30	19¼	0.77	678	1,160	0.23	830	1,420	0.42	958	1,640	0.65	1174	2,010	1.19
35	22½	1.04	580	1,570	0.31	710	1,925	0.58	820	2,220	0.87	1005	2,720	1.63
40	25¾	1.36	508	2,065	0.41	623	2,530	0.75	719	2,920	1.16	880	3,580	2.13
45	29⅞	1.75	451	2,600	0.52	553	3,185	0.96	639	3,680	1.47	783	4,510	2.70
50	32⅛	2.16	407	3,220	0.64	498	3,940	1.18	575	4,550	1.82	705	5,580	3.35
55	35⅜	2.61	369	3,890	0.77	452	4,765	1.42	522	5,500	2.19	640	6,740	4.03
60	38½	3.13	339	4,630	0.92	415	5,675	1.70	479	6,550	2.61	587	8,030	4.80
70	45	4.26	290	6,320	1.25	355	7,730	2.31	410	8,930	3.55	502	10,920	6.52
80	51⅜	5.54	254	8,230	1.64	315	10,080	3.02	359	11,630	4.65	440	14,250	8.55
90	57⅞	7.10	226	10,410	2.08	276	12,750	3.82	319	14,730	5.88	391	18,050	10.80
100	64¼	8.75	203	12,880	2.56	248	15,750	4.71	287	18,200	7.25	352	22,300	13.32
110	70¾	10.57	185	15,550	3.10	226	19,100	5.71	261	22,000	8.78	320	26,950	16.12
120	77¼	13.00	169	18,530	3.69	207	22,700	6.78	239	26,200	10.44	293	32,080	19.18
130	83½	14.85	156	21,600	4.31	192	26,450	7.93	221	30,550	12.20	271	37,410	22.40
140	90	17.20	145	25,200	5.02	177	30,850	9.24	205	35,650	14.20	251	43,700	26.10
150	96½	19.70	135	28,950	5.76	165	35,400	10.60	191	40,900	16.30	234	50,150	29.95
160	103	22.40	127	32,800	6.57	154	40,200	12.10	179	46,450	18.60	219	56,900	34.15
170	109¼	25.40	120	37,150	7.42	146	45,500	13.65	169	52,550	21.00	207	64,400	38.60
180	115¾	28.50	112	41,700	8.31	138	51,100	15.25	159	59,000	23.50	195	72,250	43.15
190	122¼	31.70	107	46,300	9.26	131	56,700	17.05	151	65,500	26.20	185	80,250	48.10
200	128½	35.30	102	51,500	10.25	125	63,100	18.85	144	72,850	29.00	176	89,200	53.30

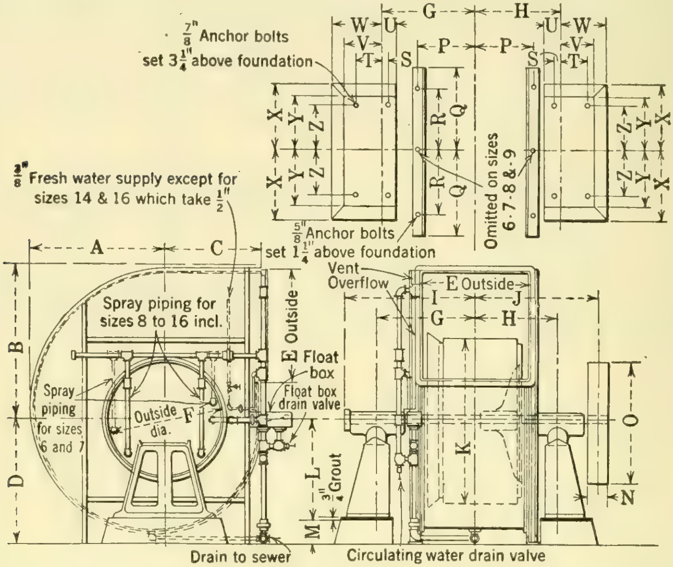
TABLE 106—*Continued*

Size	Diameter of Blast Wheel, Inches	Area of Outlet, Square Feet	2-In. Static Pressure = 1.154 Oz.			$2\frac{1}{2}$-In. Static Pressure = 1.442 Oz.			3-In. Static Pressure = 1.734 Oz.			$3\frac{1}{2}$-In. Static Pressure = 2.019 Oz.		
			R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power	R.p.m.	Volume of cubic feet per minute	Horse-power
30	$19\frac{1}{4}$	0.77	1355	2,320	1.84	1515	2,595	2.57	1660	2,840	3.38	1792	3,070	4.26
35	$22\frac{1}{2}$	1.04	1160	3,140	2.52	1295	3,510	3.48	1420	3,845	4.63	1534	4,155	5.83
40	$25\frac{3}{4}$	1.36	1018	4,135	3.28	1135	4,620	4.58	1245	5,060	6.03	1345	5,460	7.60
45	$29\frac{7}{8}$	1.75	904	5,210	4.15	1010	5,825	5.81	1108	6,375	7.63	1195	6,890	9.63
50	$32\frac{1}{8}$	2.16	814	6,440	5.15	910	7,200	7.20	996	7,880	9.45	1076	8,510	11.91
55	$35\frac{3}{8}$	2.61	738	7,780	6.19	826	8,700	8.66	904	9,530	11.38	976	10,290	14.34
60	$38\frac{1}{2}$	3.13	678	9,260	7.38	758	10,370	10.31	830	11,340	13.55	896	12,250	17.10
70	45	4.26	580	12,630	10.02	648	14,120	14.03	710	15,460	18.45	767	16,700	23.25
80	$51\frac{3}{8}$	5.54	508	16,450	13.12	568	18,400	18.40	621	20,150	24.20	672	21,750	30.50
90	$57\frac{7}{8}$	7.10	451	20,850	16.60	505	23,300	23.30	553	25,500	30.55	597	27,550	38.50
100	$64\frac{1}{4}$	8.75	406	25,750	20.48	454	28,800	28.70	497	31,530	37.70	537	34,050	47.50
110	$70\frac{3}{4}$	10.57	369	31,100	24.80	413	34,800	34.70	452	38,100	45.60	488	41,200	57.50
120	$77\frac{1}{4}$	13.00	338	37,050	29.50	378	41,400	41.30	414	45,400	54.25	447	49,000	68.40
130	$83\frac{1}{2}$	14.85	313	43,250	34.50	350	48,350	48.25	383	52,900	63.40	413	57,200	80.00
140	90	17.20	290	50,400	40.15	324	56,400	56.15	355	61,750	73.80	384	66,700	93.00
150	$96\frac{1}{2}$	19.70	270	57,900	46.10	302	64,750	64.50	331	70,900	84.70	358	76,600	106.80
160	103	22.40	253	65,700	52.60	283	73,500	73.50	310	80,400	96.60	335	86,900	121.80
170	$109\frac{1}{4}$	25.40	239	74,300	59.40	267	83,200	83.00	293	91,000	109.00	316	98,400	137.50
180	$115\frac{3}{4}$	28.50	225	83,500	66.40	251	93,400	93.00	277	102,200	122.20	298	110,400	154.00
190	$122\frac{1}{4}$	31.70	214	92,650	74.20	239	103,700	103.60	262	113,300	136.00	282	122,500	171.50
200	$128\frac{1}{2}$	35.30	204	103,000	82.00	228	115,100	114.70	250	126,100	150.80	269	136,300	190.00

Total Pressure is 126 per cent of the Rated Static Pressure.

TABLE 107

Dimensions of "ABC" Air Washing and Cooling Fans



Fan Number	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O
6	33 3/8	38 1/8	27 1/2	38	25 3/4	27 3/4	27	22 1/2	36 7/16	33 3/4	36	32 3/4	5 1/4	5 1/2	24
7	38 5/8	44 1/8	31 1/4	43	31 3/4	32 1/4	30	25 1/2	40 13/16	38 1/4	42	40 1/2	2 1/2	6 1/2	28
8	43 7/8	50 1/8	35 1/4	48	34 3/4	36 3/4	33	27 1/2	43 13/16	40 3/4	48	45	3	7 1/2	33
9	49 3/8	56 1/8	39 5/8	54	39 3/4	40 3/4	36	30 3/4	48 7/16	45 3/4	54	34 3/4	19 1/4	7 3/4	38
10	54 5/8	62 1/8	43 1/2	59	42 3/4	46 3/4	39	33	51 7/16	48 1/2	60	34 3/4	24 1/4	8 3/4	48
11	59 7/8	68 1/8	47 1/4	64	45 3/4	51 3/4	42	34 1/2	55 9/16	51	66	40 3/4	23 1/4	8 3/4	54
12	65 1/8	74 1/8	51 3/8	69	48 3/4	57 3/4	45	35 3/4	58 9/16	52 1/4	72	40 3/4	28 1/4	8 3/4	58
14	76	86 1/8	59 1/2	80	54 3/4	68 3/4	48	40 3/4	62 9/16	59 3/4	84	41 1/2	38 1/2	11 3/4	86
16	86 5/8	98 1/8	67 3/8	91	60 3/4	79 3/4	51	43 3/4	65 9/16	62 3/4	96	41 1/2	49 1/2	11 3/4	90

Fan Number	P	Q	R	S	T	U	V	W	X	Y	Z	&	Shaft at Bearing	Shaft at Wheel
6	14 1/4	22	18	1 3/4	7 1/2	5 3/4	11 1/2	13	20 1/2	19 1/4	15 1/4	1	2 15/16	2 15/16
7	17 1/4	25 1/2	21 1/2	1 3/4	9	5 3/4	13	13 1/2	22 1/4	21 3/4	17 3/4	1	3 7/16	3 7/16
8	18 7/8	28	23	1 3/4	10 1/4	5 3/4	14 3/4	15 1/4	25	24	20	1	3 7/16	3 11/16
9	21 3/8	32	27	3	10 1/8	7	14 1/8	19	28 3/4	23 7/8	19 7/8	1	3 15/16	3 15/16
10	22 7/8	35	30	3	10 1/8	7	14 1/8	20 1/2	30 1/4	23 7/8	19 7/8	1	3 15/16	4 7/16
11	24 3/8	38 1/2	34	3	10 1/8	7	14 1/8	20 1/2	30 1/4	23 7/8	19 7/8	1	4 7/16	4 7/16
12	25 7/8	41 1/2	36	3	10 1/8	7	14 1/8	21 1/2	31 1/4	23 7/8	19 7/8	1	4 7/16	4 15/16
14	29 5/8	46 3/4	40	3	10 1/8	7	14 1/8	24 1/2	34 1/4	23 7/8	19 7/8	1 1/4	4 15/16	4 15/16
16	32 5/8	55 3/4	49	3	10 1/8	7	14 1/8	27 1/2	37 1/4	23 7/8	19 1/3	1 1/2	4 15/16	5 7/16

TABLE 108

CAPACITY TABLE FOR THE "ABC" AIR WASHING AND COOLING FANS

Diameter of Fan	C.F.M.	Outlet Velocity	Static Pressure in Inches, W. C.					
			¼-in.		½-in.		¾-in.	
			B.hp.	R.p.m.	B.hp.	R.p.m.	B.hp.	R.p.m.
30	4,000	1000	.78	339	1.3	420	1.89	489
30	4,800	1200	1.11	368	1.68	446	2.33	512
30	5,600	1400	1.54	402	2.18	475	2.86	537
36	4,340	1000	.847	282	1.41	350	2.04	407
36	5,210	1200	1.21	307	1.83	372	2.54	427
36	6,080	1400	1.67	335	2.37	396	3.11	448
42	6,680	1000	1.31	341	2.18	300	3.14	349
42	8,020	1200	1.86	252	2.81	318	3.91	366
42	9,350	1400	2.56	257	3.64	339	4.78	386
48	8,040	1000	1.57	211	2.62	262	3.78	306
48	9,650	1200	2.23	230	3.39	279	4.68	320
48	11,250	1400	3.08	251	4.37	297	5.75	336
54	10,040	1000	1.96	188	3.28	233	4.74	272
54	12,050	1200	2.79	204	4.22	248	5.85	284
54	14,050	1400	3.85	223	5.47	264	7.18	298
60	11,680	1000	2.28	169	3.8	210	5.5	244
60	14,000	1200	3.24	183	4.91	223	6.81	256
60	16,350	1400	4.48	200	6.37	238	8.35	268
66	14,700	1000	2.74	154	4.57	191	6.63	222
66	16,880	1200	3.89	167	5.93	202	8.21	233
66	19,700	1400	5.40	182	7.67	216	10.10	244
72	16,000	1000	3.12	141	5.21	175	7.54	204
72	19,200	1200	4.44	153	6.74	186	9.35	213
72	22,400	1400	6.14	167	8.71	198	11.49	224
84	20,200	1000	3.94	121	6.58	150	9.52	175
84	24,240	1200	5.62	131	8.48	159	11.78	183
84	28,300	1400	7.75	143	11.00	170	14.45	192
96	25,000	1000	4.88	101	8.15	131	11.77	153
96	30,000	1200	6.94	115	10.50	139	14.60	160
96	35,000	1400	9.60	125	13.60	148	17.90	168
108	33,050	1000	6.44	94	10.75	117	15.55	136
108	39,600	1200	9.16	102	13.86	124	19.07	142
108	46,300	1400	12.70	112	18.00	132	23.2	149
120	39,100	1000	7.62	85	12.7	105	18.4	122
120	46,900	1200	10.85	92	16.4	112	22.8	128
120	54,700	1400	15.00	100	21.3	119	28.0	134
132	45,600	1000	8.88	77	14.8	96	21.45	111
132	54,700	1200	12.65	84	19.2	101	26.60	116
132	63,800	1400	17.4	91	24.80	108	32.60	122
144	52,600	1000	10.25	70	17.10	88	24.75	102
144	63,100	1200	14.60	77	22.10	93	30.70	107
144	73,600	1400	20.20	84	28.62	99	37.60	112
156	60,000	1000	11.70	66	19.50	81	28.21	94
156	72,000	1200	16.67	70	25.25	86	35.0	99
156	84,000	1400	23.00	77	32.70	92	43.0	103

These capacities are for Single Inlet Fans. At the same speeds and pressures Double Inlet Fans will deliver twice the volume of air and will require twice the amount of power for operating.

TABLE 109

Carrier Type "A" Air Washer

Dimensions and Capacities

Free Area, Square Feet	Washing Surface, Square Feet	Size of Door, Inches	Gallons per Minute: Spray	Gallons per Minute: Flooding	Gallons per Minute: Both	Water Pipes: To pump	Water Pipes: Fresh	Pump: Size	Pump: R.p.m.	Pump: Horse power: Brake	Pump: Horse power: Size of motor	Steam Pipe: 0 lb.	Steam Pipe: 5 lb.	Height: Ft.	Height: In.	Width: Ft.	Width: In.	Length: Ft.	Length: In.	Capacity in Cubic Feet of Air per Minute	Number
2.95	28.6		9	5	14	1½	¾	1½	1700	.9	2	1	1			1	5¼			1,500	1A
6.23	60.5		18	10	28	1½	¾	1½	1700	1.2	2	1½	1¼			2	9			3,100	2A
9.50	92.5		27	15	42	2	¾	1½	1700	1.5	2	2	1½			4	0¾			4,800	3A
12.8	124.0		36	20	56	2	¾	1½	1700	1.8	3	2	2			5	4½			6,400	4A
16.0	155.0		45	25	70	2	¾	1½	1700	2.1	3	2½	2			6	8			8,000	5A
19.3	187.0		54	31	85	3	¾	2	1700	2.4	3	2½	2½	4	1½	7	11¾			9,700	6A
22.6	219.0		63	36	99	3	¾	2	1700	2.7	3	3	2½			9	3½			11,300	7A
25.9	251.0		72	41	113	3	¾	2	1700	3.0	5	3	2½			10	7¼			13,000	8A
29.1	282.0		81	46	127	3	1	2½	1700	3.3	5	3	3			11	11			14,600	9A
32.4	315.0		90	52	142	3	1	2½	1700	3.6	5	3½	3			13	2½			16,200	10A
35.7	346.0		99	57	156	3	1	2½	1700	3.9	5	3½	3			14	6¼			17,800	11A
4.13	39.2	15½ × 26	11	5	16	2	¾	1½	1700	1.0	2	1¼	1			1	5¼			2,100	1B
8.71	84.5		22	10	32	2	¾	1½	1700	1.3	2	2	1½			2	9			4,400	2B
13.3	129.0		33	15	48	2	¾	1½	1700	1.6	2	2	1½			4	0¾	7	2 5/16	6,700	3B
17.9	174.0		43	20	63	2	¾	1½	1700	1.9	3	2½	2½			5	4½			9,000	4B
22.5	218.0		54	25	79	2½	¾	2	1700	2.3	3	3	2½			6	8			11,300	5B
27.1	263.0		65	31	96	3	¾	2	1700	2.6	3	3	3			7	11¾			13,600	6B
31.7	308.0		76	36	112	3	¾	2	1700	3.0	5	3	3	5	1½	9	3½			15,800	7B
36.2	351.0		87	41	128	3	¾	2½	1700	3.3	5	3½	3			10	7¼			18,100	8B
40.8	396.0		97	46	143	3	1	2½	1700	3.6	5	3½	3½			11	11			20,400	9B
45.4	440.0		108	52	160	3	1	2½	1700	4.0	5	4	3½			13	2½			22,700	10B
50.0	485.0		119	57	176	3	1	2½	1700	4.4	5	4	3½			14	6¼			25,000	11B
54.6	530.0		130	62	192	3	1	2½	1700	4.7	7½	4	4			15	10			27,300	12B
6.40	62.0		18	5	23	1½	¾	1½	1700	1.1	2	1½	1¼			1	5¼			3,200	1C
13.7	133.0	16 × 36	36	10	46	2	¾	1½	1700	1.6	2	2	2	7	1½	2	9			6,900	2C
20.9	203.0		54	15	69	2	¾	1½	1700	2.1	3	2½	2½			4	0¾			10,500	3C

28.1	273.0	16 × 36	72	20	92	3	¾	2	1700	2.6	3	3	3	7	1½	5	4½	7	2 5/16	14,100	4C
35.3	343.0		90	25	115	3	¾	2	1700	3.1	5	3½	3			6	8			17,700	5C
42.5	413.0		108	31	139	3	¾	2½	1700	3.6	5	3½	3½			7	11¾			21,300	6C
49.7	483.0		126	36	162	3	¾	2½	1700	4.1	5	4	3½			9	3½			24,900	7C
56.9	553.0		144	41	185	3	¾	2½	1700	4.6	5	4	4			10	7¼			28,500	8C
64.1	623.0		162	46	208	3	1	2½	1700	5.1	7½	4½	4			11	11			32,100	9C
71.3	693.0		180	52	232	4	1	3	1700	5.6	7½	5	4½			13	2½			35,700	10C
78.5	763.0		198	57	255	4	1	3	1700	6.1	7½	5	4½			14	6¼			39,300	11C
85.7	852.0		216	62	278	4	1	3	1700	6.5	10	5	4½			15	10			42,800	12C
92.9	902.0		234	67	301	5	1	4	1120	7.1	10	6	5			17	1¾			46,500	13C
100.0	970.0		252	73	325	5	1	4	1120	7.5	10	6	5			18	5½			50,000	14C
107.0	1040.0		270	78	348	5	1	4	1120	7.9	10	6	5			19	9¼			53,500	15C
114.0	1110.0		288	83	371	5	1¼	4	1120	8.3	10	6	6			21	1			57,000	16C
122.0	1180.0		306	88	394	5	1¼	4	1120	8.7	10	6	6			22	4½			61,000	17C
129.0	1250.0		324	93	417	5	1¼	4	1120	9.0	10	7	6			23	8¼			65,000	18C
136.0	1320.0		342	99	441	5	1¼	4	1120	9.4	10	7	6			25	0			68,000	19C
143.0	1390.0		360	104	464	5	1¼	4	1120	9.8	15	7	6			26	3¾			72,000	20C
150.0	1460.0		378	109	487	5	1¼	4	1120	10.1	15	7	6			27	7½			75,000	21C
158.0	1530.0		396	114	510	5	1¼	4	1120	10.5	15	7	7			28	11			79,000	22C
165.0	1600.0		414	120	534	6	1¼	5	1120	10.9	15	7	7			30	2¾			83,000	23C
8.9	86.0		24	5	29	1½	¾	1½	1700	1.2	2	2	1½	9	1½	1	5¼			4,400	1D
18.7	181.0		47	10	57	2	¾	1½	1700	1.8	3	2½	2½			2	9			9,400	2D
28.5	276.0		70	15	85	2½	¾	2	1700	2.5	3	3	3			4	0¾			14,300	3D
38.4	373.0		94	20	114	3	¾	2	1700	3.0	3	3½	3			5	4½			19,200	4D
48.2	468.0		117	25	142	3	¾	2½	1700	3.6	7½	3½	3½			6	8			24,100	5D
58.0	563.0		140	31	171	3	¾	2½	1700	4.2	7½	4½	4			7	11¾			29,000	6D
67.8	658.0		164	36	200	3	¾	2½	1700	4.9	7½	4½	4			9	3½			33,900	7D
77.6	754.0		187	41	228	4	¾	3	1700	5.5	7½	5	4½			10	7¼			38,800	8D
87.4	847.0		210	46	256	4	1	3	1700	6.1	10	5	5			11	11			43,700	9D
97.2	940.0		234	52	286	4	1	3	1700	6.7	10	6	5			13	2½			48,600	10D
107.0	1040.0		258	57	315	5	1	4	1120	7.2	10	6	5			14	6¼			53,500	11D
117.0	1140.0		281	62	343	5	1	4	1120	7.8	10	6	6			15	10			59,000	12D
127.0	1230.0		304	67	371	5	1	4	1120	8.3	10	6	6			17	1¾			64,000	13D
137.0	1330.0		328	73	401	5	1	4	1120	8.7	15	7	6			18	5½			69,000	14D
146.0	1420.0		352	78	430	6	1	5	1120	9.2	15	7	6			19	9¼			73,000	15D
156.0	1520.0		375	83	458	6	1¼	5	1120	9.7	15	7	7			21	1			78,000	16D
166.0	1610.0		398	88	486	6	1¼	5	1120	10.1	15	7	7			22	4½			83,000	17D
176.0	1710.0		422	93	515	6	1¼	5	1120	10.6	15	8	7			23	8¼			88,000	18D
186.0	1800.0		445	99	544	6	1¼	5	1120	11.0	15	8	7			25	0			93,000	19D

TABLE 109—*Continued*

Free Area, Square Feet	Washing Surface, Square Feet	Size of Door, Inches	Gallons per Minute			Water Pipes		Pump				Steam Pipe		Height		Width		Length		Capacity in Cubic Feet of Air per Minute	Number
			Spray	Flooding	Both	To pump	Fresh	Size	R.p.m.	Horse power: Brake	Horse power: Size of motor	0 lb.	5 lb.	Ft.	In.	Ft.	In.	Ft.	In.		
195.0	1900.0		468	104	572	6	1¼	5	1120	11.5	15	8	7			26	3¾			98,000	20D
205.0	1990.0		492	109	601	6	1¼	5	1120	11.9	15	8	8	9	1½	27	7½			103,000	21D
215.0	2090.0		515	114	629	6	1¼	5	1120	12.4	15	8	8			28	11			108,000	22D
225.0	2190.0		538	120	658	6	1¼	5	1120	12.9	15	8	8			30	2¾			113,000	23D
11.2	109.0		29	5	34	1½	¾	1½	1700	1.3	2	2	2			1	5¾			5,600	1E
23.6	229.0		58	10	68	2½	¾	1½	1700	2.0	3	3	2½			2	9½			11,800	2E
36.1	350.0		87	15	102	3	¾	2	1700	2.8	5	3½	3			4	1¼			18,100	3E
48.6	472.0		115	20	135	3	¾	2½	1700	3.5	5	4	3½			5	5			24,300	4E
61.0	592.0		144	25	169	3	¾	2½	1700	4.2	5	4½	4			6	8½			31,000	5E
73.4	712.0		173	31	204	3	¾	2½	1700	5.0	7½	5	4½			8	0¼			36,700	6E
85.8	833.0		202	36	238	4	¾	3	1700	5.7	7½	5	4½			9	4			42,900	7E
98.2	953.0		230	41	271	4	¾	3	1700	6.4	7½	6	5			10	7¾			49,100	8E
110.0	1070.0		259	46	305	5	1	4	1120	7.1	10	6	6			11	11½			55,000	9E
123.0	1190.0	16 × 36	288	52	345	5	1	4	1120	7.8	10	6	6			13	3	7	$2\frac{5}{16}$	62,000	10E
135.0	1310.0		317	57	374	5	1	4	1120	8.3	10	7	6			14	6¾			68,000	11E
148.0	1430.0		346	62	408	5	1	4	1120	8.9	10	7	6	11	1¾	15	10½			74,000	12E
160.0	1550.0		375	67	442	5	1	4	1120	9.4	10	7	7			17	2¼			80,000	13E
173.0	1680.0		404	73	477	5	1	4	1120	10.0	15	7	7			18	6			87,000	14E
185.0	1800.0		432	78	510	5	1	4	1120	10.5	15	8	7			19	9¾			93,000	15E
198.0	1920.0		462	83	545	6	1¼	5	1120	11.0	15	8	7			21	1½			99,000	16E
210.0	2040.0		490	88	578	6	1¼	5	1120	11.5	15	8	8			22	5			105,000	17E
222.0	2160.0		518	93	611	6	1¼	5	1120	12.1	15	8	8			23	8¾			111,000	18E
235.0	2280.0		547	99	646	6	1¼	5	1120	12.6	15	10	8			25	0½			118,000	19E
241.0	2400.0		576	104	680	6	1¼	5	1120	13.2	15	10	8			26	4½			124,000	20E
260.0	2520.0		605	109	714	6	1¼	5	1120	13.7	15	10	8			27	8			130,000	21E
273.0	2650.0		634	114	748	6	1¼	5	1120	14.2	15	10	10			28	11½			137,000	22E
285.0	2770.0		663	120	783	6	1¼	5	1120	14.7	20	10	10			30	3½			143,000	23E

CHAPTER XVI

MISCELLANEOUS APPLICATIONS OF REFRIGERATION

Some of the applications of refrigeration are very special, and it seems best to bring these all together into one chapter. In some of the following there is no established method of solution, while in others, for example, in the case of the problem in shaft-sinking, refrigeration has been abandoned, practically, as a modern method, but some of the devices outlined, however, may be useful in solving other problems not mentioned specifically in the text.

SKATING RINKS

The skating rink is one of the popular uses to which mechanical refrigeration has been applied. Although it would seem at first that it is similar in most respects to other applications, yet in reality it is very different, in fact unique, in the details of design and operation. Of course, there is the factor of the making of ice, and yet it is entirely different from commercial ice making, inasmuch as skating rinks require an ice surface of particular quality and temperature and nothing else. It has been found that the best surface for skating is one that is very nearly at 32 deg. F., but which is not wet on the surface and has at the same time a temperature of from 22 to 24 deg. F. near the refrigerating coils. A room temperature of from 48 to 50 degrees is comfortable and is the most desirable for skaters.

Design of Skating Rinks.—The skating floor should be as solid as possible. Unless concrete is used it is well to have about 10 to 12 in. of sand on which are laid stringers on about 24 to 36-in. centers to which the piping is attached. The piping can be either 1 in. or $1\frac{1}{4}$ in., erected on from 3 to 6-in. centers, but 4 to $4\frac{1}{2}$-in. centers will give the best results, as there is less likelihood of soft places between the pipes if the larger spacing is not used. The brine system is the one used almost entirely because of the freedom from variable temperatures, and because of the ability with brine to store up refrigeration in a brine tank for the occasion of a peak load which is almost inevitable in the case of the

skating rink. A brine temperature of about 16 to 18 deg. F. gives the most satisfactory results, as it has been found that if the ice is too cold, and hard, it does not wear well but forms snow easily—a condition that does not exist with the ice at or near 32 degrees.

The brine lines should be arranged so as to be readily accessible. There is nothing standard in regard to the supports for the pipes. The Vancouver, B. C., rink is built on lumber, while the Duquesne Garden was built on a wooden floor with some three or four feet of open space beneath it. The Winter Garden, Pittsburgh, is laid entirely, and the Seattle rink, partly, on concrete. The Chicago Arena, the Cleveland, the Portland (Oregon), and the Victoria rinks were laid directly on the floor. The essential is that the floor be level. Insulation of the floor is considered unnecessary and simply an added expense.

There is no standard method of piping. Some rinks are designed for a reversal of brine flow and others are not. As a rule the brine headers run the long way of the rink and the pipes are arranged to run across the short dimension. Some engineers specify that the piping shall be laid out so that each separate circuit is of equal length in order that the regulation of flow can be obtained without throttling the shorter circuits. The amount of brine flow varies considerably also, but a brine temperature rise of from two to four degrees only, should be permitted, and the temperature during the operation of the rink should be approximately from 16 to 18 degrees. It is essential that brine piping be laid out so that air pockets cannot be formed and so that the air be carried upwards to a point where air cocks can be placed. The resistance to the flow of the brine should not be greater than 4 to 6 lb. per sq. in. in order to keep the pumping cost of operation as low as practical. A ratio of 1 gal. of brine per minute to 23 sq. ft. of floor area was used with success in the Chicago Arena.

As regards methods of calculation of the tonnage required, it will be more satisfactory to refer to the table showing what has been done before in the case of rinks that have been successful. Some tests by D. H. Scott at the Cleveland rink showed that a heating effect of 21,000,000 B.t.u. per day prevailed at the time of the test. Of this amount it was estimated that 18 per cent was due to direct sunlight, 12 per cent to heat leakage through the walls and the roof, 3 per cent to air changes, 20 per cent to requirements in freezing the ice, 9 per cent to a special temperature control used at this rink, 8 per cent to the heat evolved by the skaters, 1 per cent to the electric illumination, leaving 29 per cent unaccounted for. It would seem that the loss due to infiltration would be more than 3 per cent, and some heat would be given by the ground. As the skating season extends from the fall through the

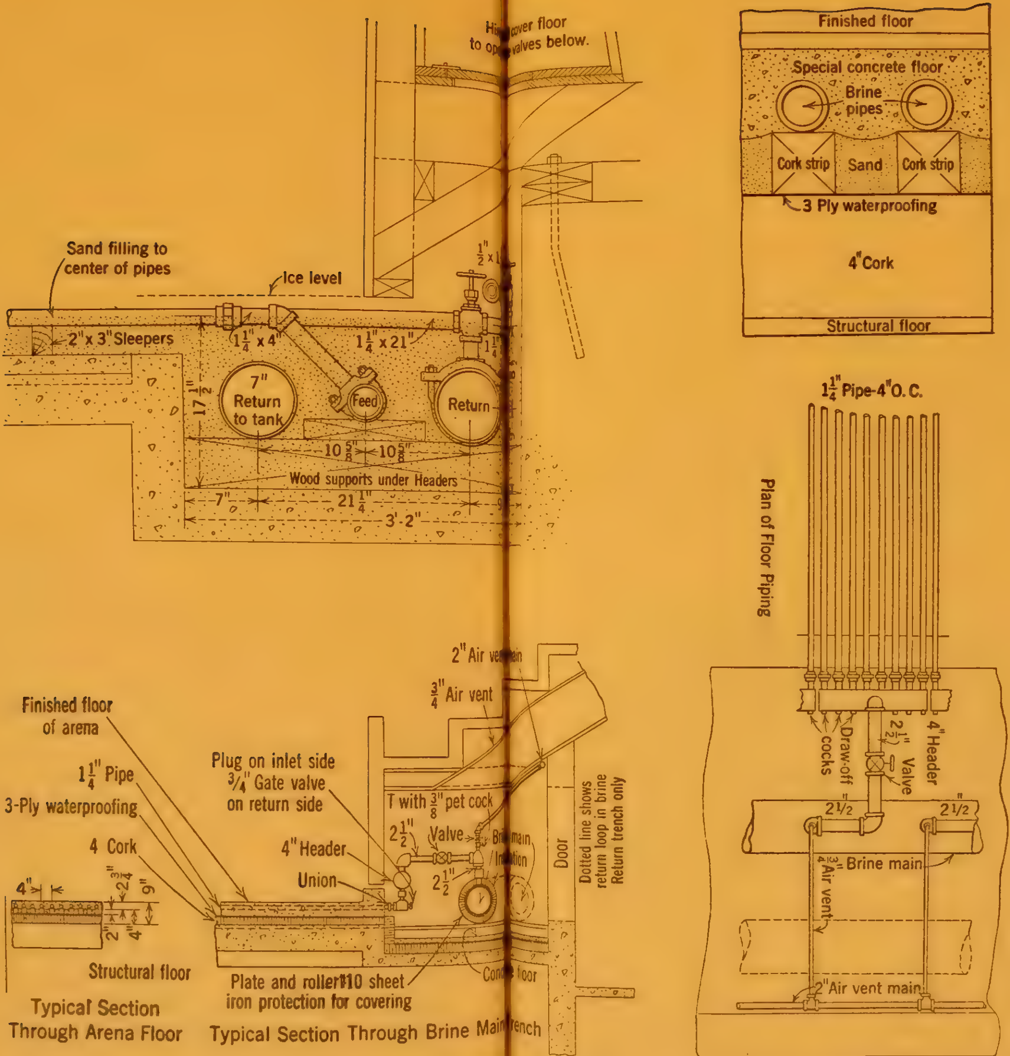


FIG. 339.—Skating Rink Details.

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winter into the spring, it is out of the question to make any careful analysis of the refrigeration requirements, and it seems much better to be governed by designs in successful operation. It has been usual to

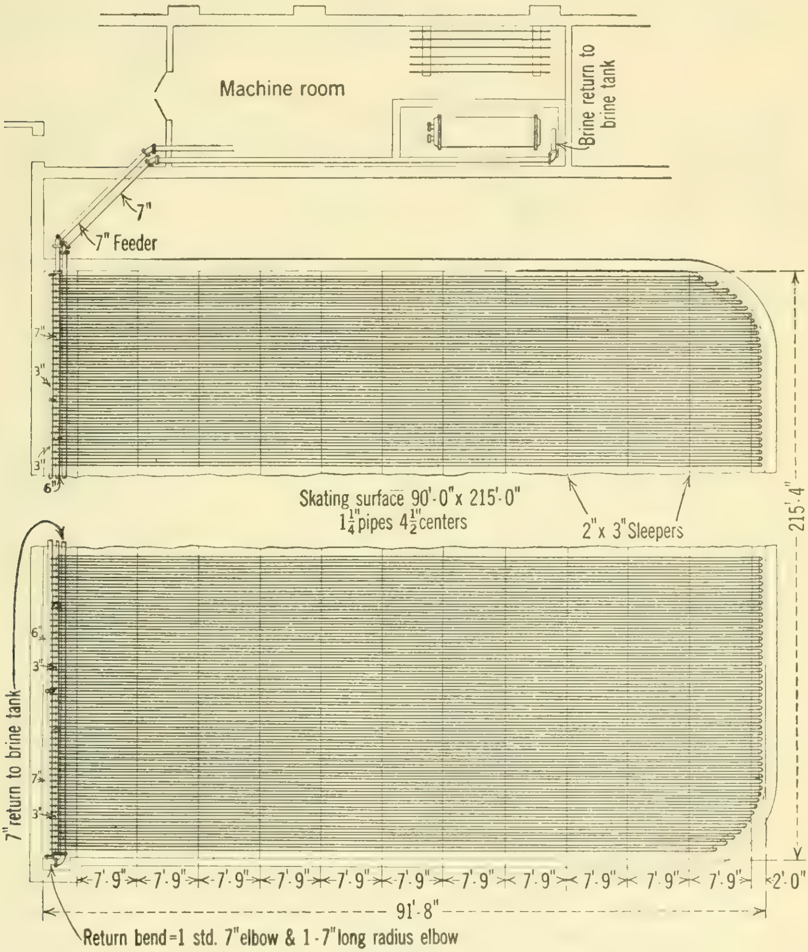


Fig. 338.—Skating Rinks.

design the piping so that the floor may be used for other purposes in the summer time.

The average cost of rinks has been $1.40 per sq. ft. of ice surface. The usual allowance is 20 sq. ft. of ice surface per skater, which is considered liberal, and 12 sq. ft. is considered to give a crowded condition. Figures 338 and 339 give typical arrangements of brine piping.

TABLE 110

	Ice Surface, Square Feet	Feet of 1-In. Pipe	Feet of $1\frac{1}{4}$-In. Pipe	Linear Feet per Square Foot of Floor	System
Madison Square Garden	15,820 (85×186)	....	53,000	3.35	Brine
Pittsburgh...........	15,750	72,000	....	4.57	Direct exp.
St. Nicholas..........	14,400	....	44,000	3.06	Brine
Chicago (arena).......	33,900	106,200	....	3.13	Brine
Berlin................	26,900	....	82,000	3.05	Brine
Washington, D. C......	23,600	....	96,000	4.06	Direct exp.
Duluth, Mich..........	19,350 (90×215)	....	52,000	2.69	Brine

	Ice Surface, Square Feet	Tonnage	Square Feet of Ice Surface per Ton
Duquesne Garden....	23,600 (90×262)	135	175
Brooklyn...........	15,285	80	191
St. Nicholas.........	14,400	80	180
Washington, D. C....	23,600	100	236
Spokane.............	16,000	65	246
Toronto.............	16,000	100	160
Cleveland............	20,500 (86×240)	130	158
Chicago (arena)......	33,925 (115×295)	215	158
Portland, Oregon.....	27,300 (85×321)	175	156

THE CHILLING OF CASTINGS

In the chilling of castings, such as in the manufacture of plowshares and mold boards, one of the objects of the treatment is to secure a uniform glass hardness. The chilling may be accomplished by the use of calcium chloride brine in the case of iron castings, but a quenching oil is used in the case of steel manufacture. It is understood that if water were used in either case the heat of the metal would generate considerable steam, and while the greater part of this would relieve itself readily, yet some would cling to the surface of the metal and form a surface film of gas. The action of such a surface film is to cause resistance to the transfer of heat, and the metal would be cooled non-uniformly and soft

and hard spots would be formed. The action of brine is different as the gas film does not seem to be so marked as in the case of the water. If oil is used a slower cooling results because of the lower value of the coefficient of heat transfer from the metal to the oil than from the metal to the brine. Experience, however, has shown that sodium chloride only

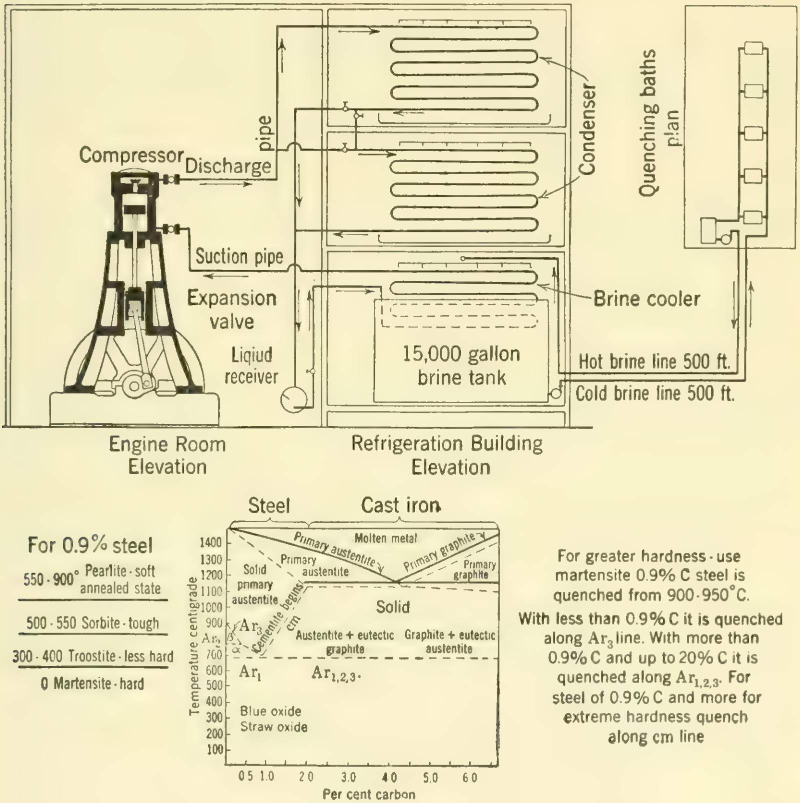


FIG. 340.—The Chilling of Cast Iron.

is the salt giving the required glass hardness expected in the manufacture of plowshares. The following is taken from the John Deere Plow Company records:

Problem.—The daily run consists of 45,500 lb. of castings which enters the brine bath at 1650 deg. F. and is cooled to 50 degrees (in 9 hours). The brine is to to be cooled to 45 deg. F. and is to be heated 10 degrees only. Find the amount of refrigeration required and the amount of brine to be circulated per minute.

The specific heat of the casting is to be taken as 0.169, and of the brine of a density of 1.115 as 0.843. The refrigeration required is

$$\frac{45500}{9} \times (1650 - 50) \times 0.169 = 1{,}367{,}000 \text{ B.t.u. per hour.}$$

$$= 113.9 \text{ tons}$$

Adding 20 per cent for unavoidable losses $= 22.8$

$$= 136.7 \text{ tons}$$

Referring to Fig. 340 it will be seen that the brine lines are quite long so that a liberal factor needs to be used for the losses. The brine required per minute will be:

$$\frac{1{,}640{,}000}{60} = M \times 0.843 \times (55 - 45).$$

$$M = 3242 \text{ lb. per minute.}$$

Taking the density of the brine at 1.115, the number of gallons to be circulated per minute will be 349.

THE POETSCH PROCESS

A construction process used some years ago, but which has been neglected of late years, is the freezing process of sinking vertical shafts in aqueous soils and quicksand. The essential feature of the process is to sink pipes for brine or direct expansion sufficiently close together to enable the water in the coil to be frozen solid. For this purpose brine at about -15 deg. F. is taken and the pipes are located about 3 ft. apart.

An estimate of the refrigeration required can be made when the details are known. Frequently the specific gravity of the dry solids is taken at 1.8, the specific heat at 0.2, and the average water content in the earth at 25 lb. per cu. ft. The frozen ring may extend $1\frac{1}{2}$ ft. outside, and 3 ft. inside the center of the line of pipes, but the ground is cooled 5 ft. beyond the frozen zone. For calculating the time of freezing, an average value of 85 B.t.u. per sq. ft. of pipe surface is recommended, and 33 per cent is added to the total calculated refrigeration for losses. The total time of freezing usually is from 5 to 6 weeks.

THE PRODUCTION OF VERY LOW TEMPERATURES

The manufacture of liquid air cannot be said to be an application of refrigeration, although at times some use of refrigerating processes is made during a part of the course of operations. For example, in one plant of the Linde Air Products Company, which has a capacity of 55,000 cu. ft. of free air per hour, a $3\frac{1}{2}$-in. by 10-in. CO_2 compressor is

used in the fourth stage after-cooler in order to separate out, by cooling the air to − 20 deg. F., as much moisture as possible at this point rather than to permit it to continue on to the expansion valve where frost would accumulate and interfere with the continuity of action. Liquid air may be produced by two principal methods: the Linde and the Claude processes.

In the Linde process the atmospheric air is purified by passing it through a lime scrubber in order to reduce the carbon dioxide content in the air to 0.003 per cent, which would also freeze on the throttle valve were it permitted to continue in the liquefaction process, and the air is then compressed. In some of the plants 55 per cent of the total capacity of the plant is compressed in four stages to 2000 lb. per sq. in., and 45 per cent is compressed to 70 lb. gage, as this ratio appears to work satisfactorily. Cylinder lubrication is secured by means of a soap solution. The depression of the temperature at the throttle valve is obtained by means of the Joule-Thompson effect,[1] and by a very carefully designed counterflow exchanger and rectification column (Fig. 341). The depression of the temperature being cumulative, the formation of liquid air finally commences at the throttle valve and floods the lower coils (the vaporizer), and by boiling performs the function of refrigerating the compressed gas in these coils to the point of liquefaction. The liquid air passes upward through d_2 to the top of the rectification column where it trickles down through the screenings, etc., of the column. The vaporized liquid from the vaporizer passes counter-current to the descending liquid air, and in due time an equilibrium is established with a differential temperature of about 20 deg. F. (− 194 deg. C. at the top and − 183 deg. C. at the bottom). As nitrogen boils at − 196 deg. C. and oxygen at − 183 deg. C., the result of this process is that the nitrogen gas, with 7 per cent oxygen, is formed at the top and oxygen, with 0.8 per cent nitrogen gas at the bottom. The remainder of the apparatus is designed to provide the counterflow required to cool the entering compressed gas and to heat the extremely cold gases leaving the coils.

The result of the Linde process is the production of nearly pure oxygen, but the nitrogen has about 7 per cent of oxygen present. Bayley's

[1] The Joule-Thompson effect for air is about 0.5 deg. F. per atmosphere drop of pressure during throttling. The depression of temperature (in deg. C.) is given by the expression

$$t_1 - t_2 = 0.276(p_1 - p_2)\left(\frac{273}{T}\right)^2$$

where p_1 and p_2 are the pressures before and after throttling, in atmospheres, and T is the initial temperature in deg. C. abs.

experiments in 1900 proved that this percentage of oxygen could not be decreased when using the rectification column.

The Claude process uses a combination of an expansion engine for two-thirds of the gas and a throttling valve for the remaining one-third. The gas leaving the expansion engine is at about four atmospheres and is at or near the liquefaction temperature point, whereas the air which arrives at the expansion valve is liquid at about 33 atmospheres. A selective system of rectification is employed which is rendered possible by a preliminary partial separation of nitrogen and oxygen and a combined system of heat interchange and drying which eliminates the necessity of a chemical removal of moisture from the air. Figure 341

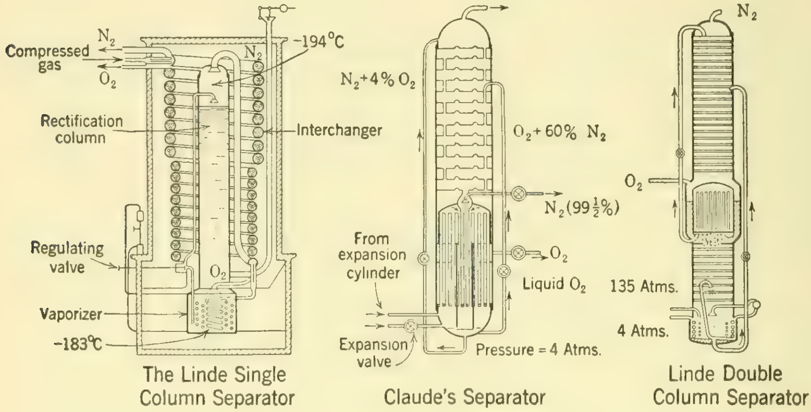


FIG. 341.—The Production of Very Low Temperatures.

shows the Claude system details to some extent. The power required in liquid air plants is approximately 30 to 33 b.hp. per 1000 cu. ft. of oxygen per hour.

THE ICE CREAM AND MILK INDUSTRY

Before the general use of mechanical refrigeration, the manufacture of ice cream and the pasteurizing of milk and cream were limited to the use of ice and salt for cooling. The result of the introduction of mechanical refrigeration has been a very rapid extension of the industry during the last twenty years. The ice cream manufactured previous to 1900 in the United States is estimated as at about 25 million gallons yearly, with few concerns of capacities greater than 100,000 gallons, whereas the capacity in 1920 was 260 million gallons yearly, and a number of

firms had outputs greater than a million gallons. The number of refrigerating machines and their capacities had doubled from 1919 to 1922. The advance in the use of mechanical refrigeration has not been so rapid in the dairy plants. The listed plants were of 9300 tons of refrigeration in 1904, 16,800 tons in 1914, and 46,400 tons in 1922, with an average tonnage per machine of 13.5 tons.

Pasteurizing of Milk.—Quick cooling of milk is an absolute necessity when the milk is not consumed immediately. Referring to Table 111

TABLE 111

EFFECT OF TIME AND TEMPERATURE ON THE GROWTH OF BACTERIA IN MILK

Temperature	24 Hours	48 Hours	96 Hours	168 Hours
32 deg. F. (0 deg. C.)	2,400	2,100	1,850	1,400
	30,000	**27,000**	**24,000**	**19,000**
39 deg. F. (4 deg. C.)	2,500	3,600	218,000	4,200,000
	38,000	**56,000**	**4,300,000**	**38,000,000**
42 deg. F. (5 deg. C.)	2,600	3,600	400,000	
	43,000	**210,000**	**5,760,000**	
46 deg. F. (6 deg. C.)	3,100	12,000	1,480,000	
	42,000	**360,000**	**12,200,000**	
50 deg. F. (10 deg. C.)	11,600	540,000		
	89,000	**1,940,000**		
55 deg. F. (13 deg. C.)	18,800	3,400,000		
	187,000	**38,000,000**		
60 deg. F. (16 deg. C.)	180,000	28,000,000		
	900,000	**168,000,000**		
68 deg. F. (20 deg. C.)	450,000	25,000,000,000		
	4,000,000	**25,000,000,000**		
86 deg. F. (30 deg. C.)	1,400,000,000			
	14,000,000,000			
94 deg. F. (35 deg. C.)	25,000,000,000			
	25,000,000,000			

good quality milk held at 42 deg. F. for 24 hours showed 2400 bacilli, whereas if held for the same time at 86 degrees the same cubic centimeter of milk would contain 1,400,000,000 bacilli. When milk is consumed in cities and large towns it frequently is between 36 and 48 hours old when delivered to the consumer, and it is likely to be from 48 to 60 hours old when it is consumed. On the other hand it has been found that milk cannot be frozen[2] with satisfactory results. It is evident, then,

[2] In Bulletin No. 98, United States Department of Agriculture (By J. T. Bowen), it says:

"According to Kasdorf, when raw milk was partly frozen at a temperature of

that raw milk is dangerous for all purposes other than for cooking except under the most sanitary conditions of production and bottling unless some very drastic method of curtailment or sterilization of the bacilli is possible. The present method is to "pasteurize" the milk (and cream) by bringing the temperature up to a point which will kill them or stunt their growth. This has been found to be achieved with satisfaction by, first, heating to about 165 deg. F. for a half-minute (the flash minute), and, second, of heating to only 140 degrees[3] and holding the temperature for 30 minutes (Fig. 342). The latter is the more usual method, as there is less heat to be removed from the milk and less likelihood of a scorched taste to the milk. After passing through the pasteurizing process the milk or cream must be cooled quickly to 40 deg. F. This is done by the so-called Baudalot cooler, an atmospheric type of cooler, or the regenerator type of cooler (Fig. 345), which permits a counterflow of the cold raw milk and the hot pasteurized milk. In the Baudalot cooler (Fig. 344), the first part of the cooling is done

10.5 deg. F., in the ordinary container, during transportation it was found that ice first formed around the sides and at the bottom of the cans; the central core contained most of the casein, sugar, and other mineral ingredients, while most of the fat was found in the top layer of the liquid portion.

"When milk which has been frozen gradually, without agitation, is thawed out clots will be found floating in the liquid (composed mostly of albumen and fat, which may be dissolved by cooking); on the other hand, if the milk is preserved in a frozen condition for three or four weeks, these clots will be very hard to dissolve, and the difficulty experienced in dissolving them increases in proportion to the length of time the milk is preserved in a frozen state. For this reason the freezing of milk for the purpose of transportation has so far been very little practiced.

"If the milk is held at 32 deg. F. for a few days some type of bacteria will grow and multiply slowly. With a good quality of milk, i.e., that containing few bacteria, it may take weeks or even months for them to gain great headway. What few bacteria develop at low temperatures are of different species from those ordinarily found at the higher temperatures, and they may produce marked changes in the chemical composition of the milk without specially changing its appearance. Consequently, it is unsafe to assume that milk which has been held for several days at a low temperature is in good condition. According to Pennington, milk exposed continually to a temperature of 29 to 32 deg. F. causes after a lapse of from 7 to 21 days the formation of small ice crystals which gradually increase until the milk is filled with them, and there may be an adherent layer on the walls of the vessel. The milk does not freeze solid. In spite of the fact that the milk was a semi-solid mass of ice crystals, an enormous increase of bacterial content took place. Though the bacterial content was numerically in the *hundreds of millions* per cubic centimeter, there was neither taste nor odor to indicate that such was the case. Neither did the milk curdle when heated, and the unfitness of the milk for household purposes would not ordinarily be detected until the lactic acid bacteria decreased in numbers and the putrefractive bacteria began to develop."

[3] Tests at Macdonald College (Ag. Gazette, Canada, 1917), claim that 152 deg. F. is required to make coli bacteria non-virulent.

with water, and the brine performs the cooling from about 80 degrees to the final temperature of 40 degrees. Too low a final temperature is not desired as the milk is likely to foam during bottling. After bottling, the cases of bottles are stored in a cooler room in order to hold the temperature until such time as delivery takes place.

The problem of refrigeration in the cooling of milk and cream is very simple. The problem is one of cooling a liquid only—from about 80 deg. to 40 deg. F. The only distinctive point is that the milk usually is a short period load, that is, it consumes some 3–5 hours per day and usually is a decided " peak " load. There are in most dairies other

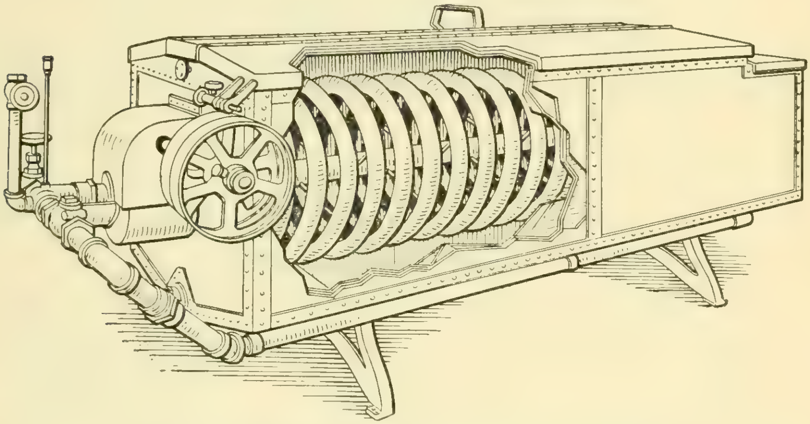


FIG. 342.—The Pasteurizing of Milk.

refrigerating needs in the same plant: for example, milk holding rooms, ice making, etc. The design could be worked out so as to carry the ice making and the storage room by one machine and the milk cooling by another. Frequently, however, the problem works out nicely by the combination with a brine tank, and by cooling the brine down to 10 deg. F. or lower sufficient refrigeration is stored up so that a smaller compressor can be used, and the machine may be operated for 8 to 12 or more hours out of the 24. Such a brine storage is well illustrated by the following problem:

Problem.—Find the refrigeration required to cool 5000 gal. of milk in three (3) hours from 75 to 40 deg. F. by the use of brine. Also find the size of brine storage required to enable the compressor to be operated for ten hours with the pasteurizer in use three (3) hours only. Take an average value of 0.9 (Fig. 343) for the specific heat of the milk and the specific gravity of the milk at 1.032 (Table 112).

The refrigeration required is

$$M \times c \times (t_2 - t_1) = 5000 \times 8\tfrac{1}{3} \times 1.032 \times 35 \times 0.9 = 1{,}355{,}000 \text{ B.t.u.}$$

$$\text{Adding 10 per cent for losses} \qquad = \quad 135{,}500 \text{ B.t.u.}$$

$$= 1{,}490{,}500 \text{ B.t.u.}$$

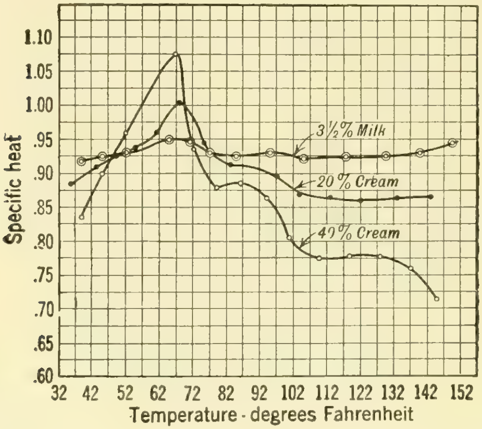


Fig. 343.—The Specific Heat of Milk and Cream.

TABLE 112

Specific Gravity of Milk and Cream Corresponding to Various Percentages of Butter Fat at 68 Deg. F.

Percentage of Fat	Specific Gravity	Percentage of Fat	Specific Gravity	Percentage of Fat	Specific Gravity	Percentage of Fat	Specific Gravity
0.025	1.037	11	1.024	21	1.012	31	1.003
1	1.036	12	1.022	22	1.011	32	1.002
2	1.035	13	1.020	23	1.010	33	1.001
3	1.034	14	1.019	24	1.009	34	1.000
4	1.032	15	1.018	25	1.008	35	.999
5	1.031	16	1.017	26	1.008	36	.999
6	1.030	17	1.016	27	1.007	37	.998
7	1.029	18	1.015	28	1.006	38	.997
8	1.027	19	1.014	29	1.005	39	.996
9	1.026	20	1.013	30	1.004	40	.995
10	1.025						

Therefore the capacity of the compressor is 1,490,500 ÷ (10 × 12,000) = 12.4 tons. If such a sized compressor was in use the brine storage would have to carry what the machine could not deliver during the 3 hours that the milk is pasteurized for

$$1{,}490{,}500 - (12.4 \times 3 \times 12{,}000) = 1{,}044{,}000 \text{ B.t.u.}$$

If brine at zero degrees is carried at the time that the milk cooling began and it was permitted to allow the brine to rise to 20 degrees then the average specific heat would be (for calcium chloride) 0.71 for 1.2 specific gravity, and the volume required would be:

$$Q = M \times c \times (t_2 - t_1)$$

$$1{,}044{,}000 = M \times 0.71 \times (20 - 0).$$

$$M_{\text{lb. of brine}} = 1{,}044{,}000 \div 14.2 = 73{,}500 \text{ lb.}$$

$$\text{Volume of brine} = 73{,}500 \div (62.5 \times 1.2) = 980 \text{ cu. ft.}$$

To store this volume of brine a cylindrical tank 11 ft. 0 in. diameter and 13 ft. 1 in. high would give the volume required, but for safety one 11 ft. diameter by 14 ft. high should be used.

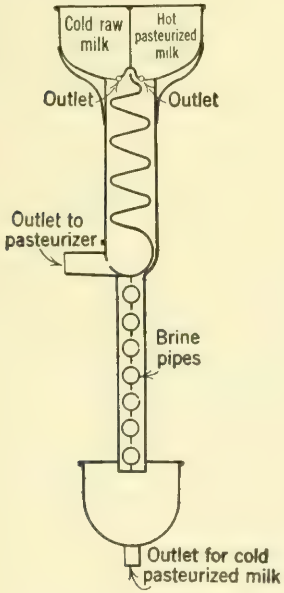


FIG. 344.—The Baudalot Milk Cooler.

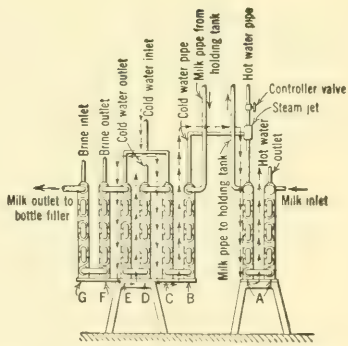
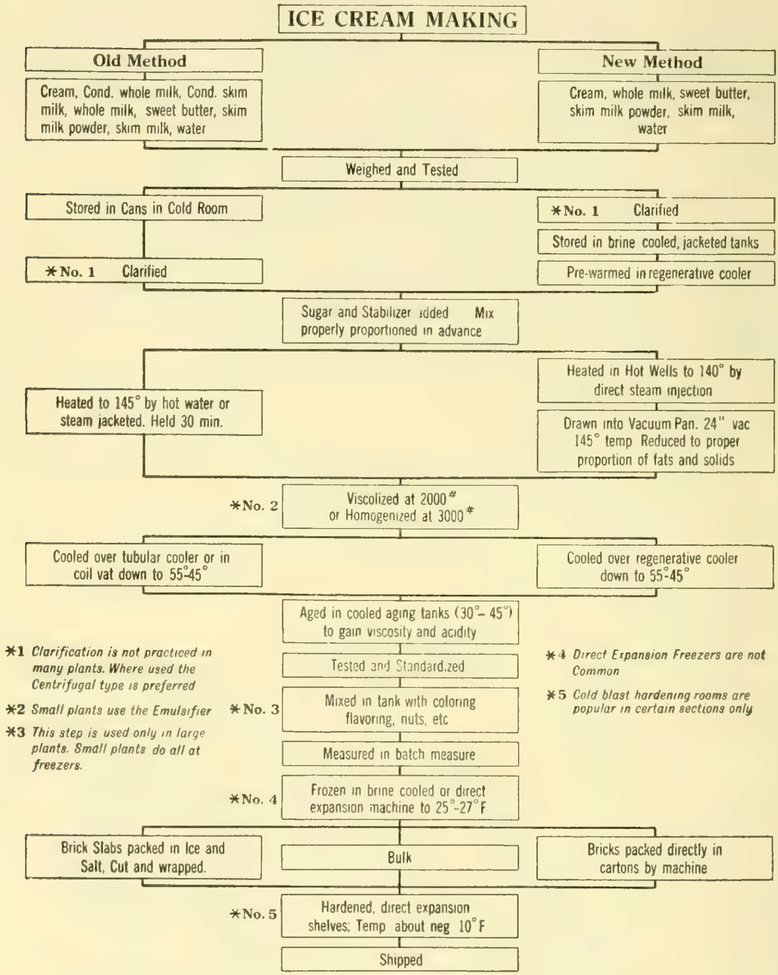


FIG. 345.—The Double Pipe Pasteurizer and Cooler.

The ice required in a milk plant does not have to be transparent, nor does it have to be any particular size or shape. Sometimes the brine tank can be utilized to manufacture ice, and 50 lb. cans will freeze in the 10 hours of operation, but the additional load should be allowed

for, using the ratio of about 200 B.t.u. of refrigeration per 1 lb. of ice manufactured.

TABLE 113.



The additional refrigeration consists of the storage room loads. When the milk is placed in the cooler it can be chilled somewhat, but not very much unless the room is heavily piped and good circulation of the air is possible. The main desire is to keep the commodity at approximately 40 degrees until the time arrives for loading on the delivery wagons. The milk bottles are placed in cans usually $18\frac{1}{2}$ in. by $14\frac{1}{2}$ in.

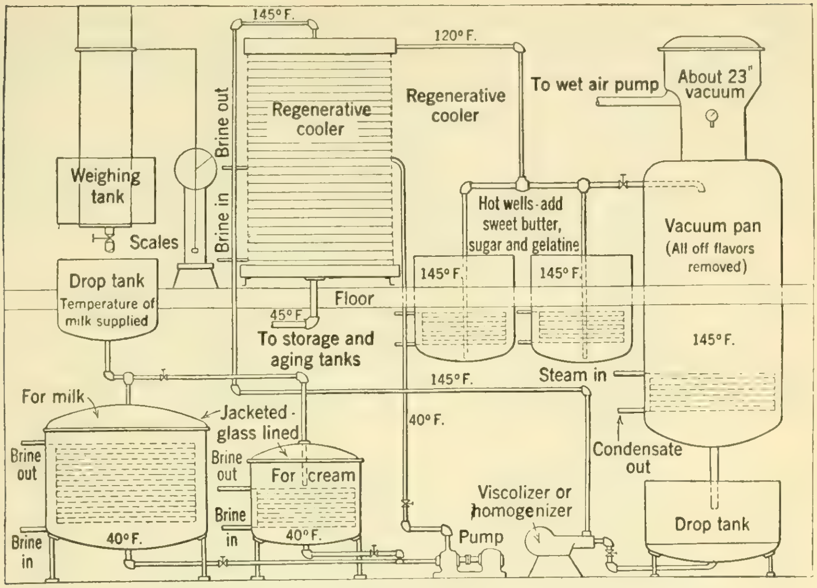


FIG. 346.—The Manufacture of Ice Cream.

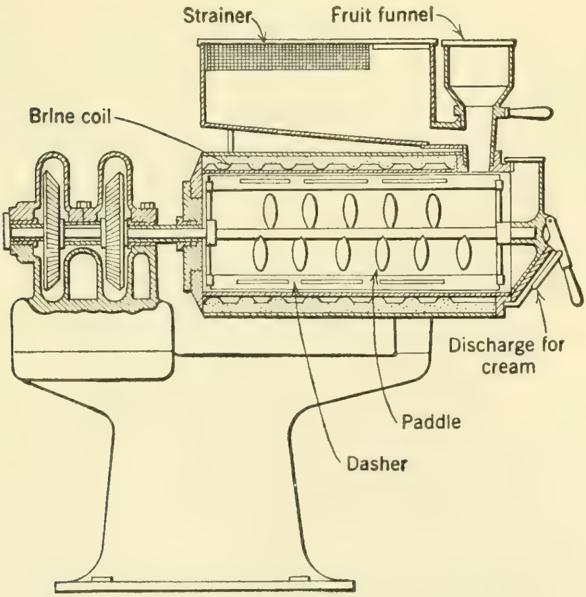


FIG. 347.—The Power Driven Ice Cream Freezer.

by 10 in. high, which hold 12 qts. or 20 pints. The weight of the case is about 14 lb., the weight of 12 one-quart bottles is 22 lb., and the weight of the milk in 12 bottles is 24 lb., making a total of about 60 lb. The

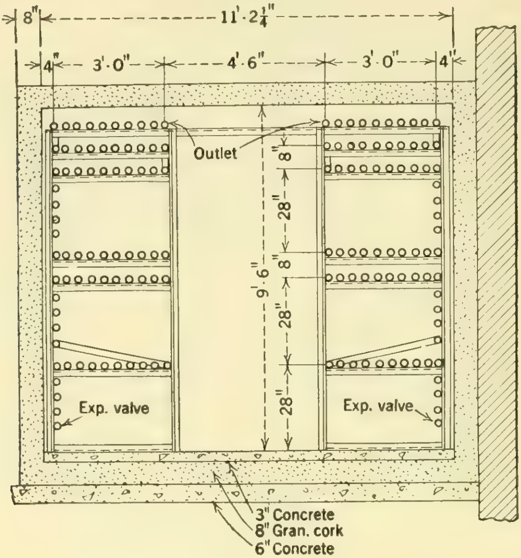


FIG. 348.—Sharp Freezer for Ice Cream.

specific heat of glass is 0.2 and of yellow pine is about 0.6. The glass milk bottle is 4 in. diameter by $9\frac{3}{8}$ in. high for quarts and $3\frac{1}{8}$ in. diameter by $7\frac{1}{4}$ in. high for pints. These last constants may be used in calculating the storage room load.

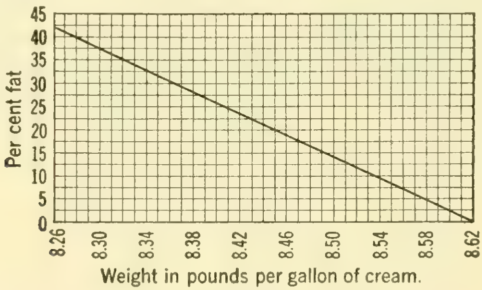


FIG. 349.—Weight of Cream.

Ice Cream Manufacture.—Although the process of the manufacture of ice cream is somewhat similar to that of ice making, yet the details are much more complicated and numerous. Table 113 shows the process,

and Fig. 346 gives an idea of the steps in preparing the mix before the freezing process. In brief, the freezing is accomplished by means of power-driven horizontal freezers only (Fig. 347). These may be 40 to 100-qt. capacity, and may be brine cooled with brine from 0 to 5 or 10 deg. F., or by means of direct expansion, using pressures from 15 to 20 lb. gage, but brine is much more easily controlled. The mix enters the freezer at about 40 to 45 degrees and is beaten by the paddles, absorbs

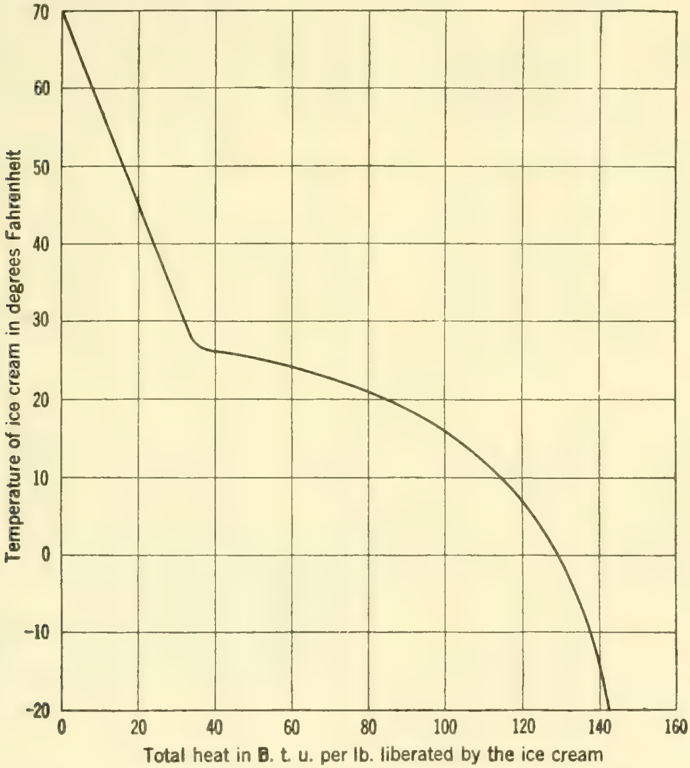


Fig. 350.—Total Heat of Ice Cream.

air, and during the freezing process at 29 to 27 deg. F. swells (acquires an *overrun*) from 50 to 100 per cent (Fig. 352), depending on the operating care during the freezing. The time of freezing is from 10 to 15 minutes, and when removed from the freezer the mix has a consistency of a viscous oil, or a heavy syrup, and is packed in large cylindrical containers of $2\frac{1}{2}$ to 5 gal. capacity, after which they are placed in hardening rooms at 0 to − 10 degrees for the rest of the freezing process. The hardening room (Fig. 348) is usually constructed with the piping

arranged as shelving in the proportion of 1 ft. of $1\frac{1}{4}$-in. pipe per $\frac{3}{4}$ cu. ft. of room space. Still air is usually employed, with an allowance of about 150 to 400 cu. ft. per 100 gal. ice cream capacity, and a corresponding floor area of 15 to 35 sq. ft. The time of storage is usually from $1\frac{1}{2}$ to 3 days.

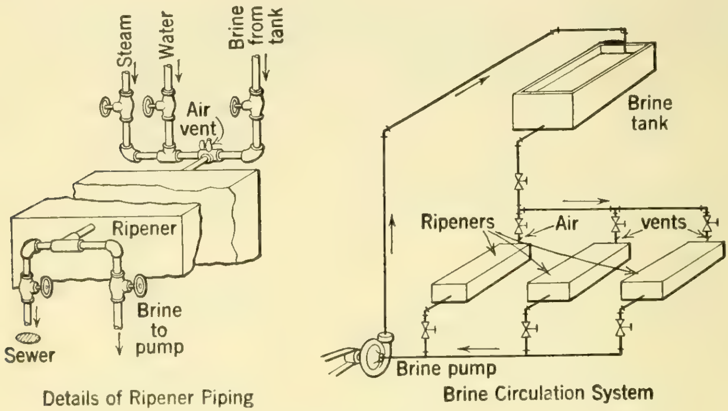


FIG. 351.—Brine Circulation in a Creamery.

The refrigeration required cannot be more than an approximation as ice cream is more like a frozen custard or pudding than anything else, and in consequence the ingredients vary greatly. It has been the custom to assume that the total amount of heat absorbed by the brine during the freezing process took place at constant temperature in the freezer,

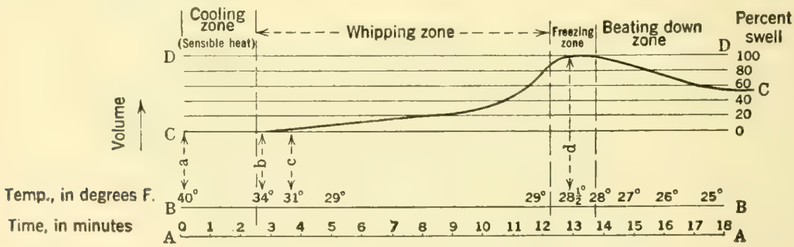


FIG. 352.—The Swell of Ice Cream.

but according to H. F. Zoller[4] careful tests indicate that for a standard vanilla ice cream of 10 per cent milk fat, 14 per cent cane sugar, $10\frac{1}{2}$ per cent milk solids not fat, and 0.5 per cent gelatine the total heat curve is as shown in Fig. 350. Here, freezing of the water content starts at

[4] H. F. Zoller, A. S. R. E., 1924, Freezing and Hardening of Ice Cream.

27 degrees and continues down to -20 degrees. The heat removed per pound of the mix by the brine or the ammonia in cooling from one

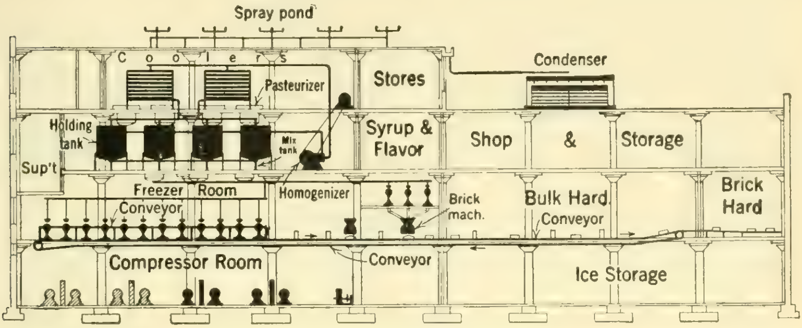


FIG. 353.—Ice Cream Manufacture.

temperature to another is found by subtracting the values of the *ordinates* of these points. For example, if the mix enters the freezer at

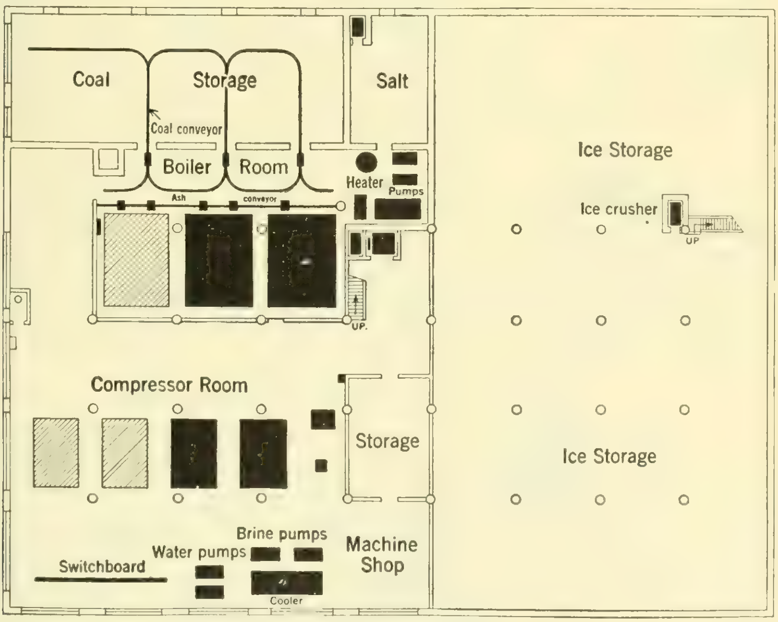


FIG. 354.—Ice Cream Manufacture.

50 degrees and leaves at 21 degrees, the heat absorbed by the refrigerant is $80 - 16 = 64$ B.t.u. per lb. If the half-frozen cream is now

placed in the sharp freezer and is lowered to 0 degrees, then the heat removed in the hardening room is 130 − 80 = 50 B.t.u. per lb.

Problem.—1000 gal. of ice cream is to be manufactured per day. The swell will be assumed to be 60 per cent. The mix enters the freezers at 40 degrees and the ice cream is kept stored in a hardening room kept at 0 deg. F.

If 1000 gal. is the amount of the finished product, then the amount of the mix is 1000 ÷ 1.6 = 625 gals. If the specific gravity of the mix can be taken as 1.10, then the weight per gallon would be 5.7 + lb. (of the product) and the total weight would be 1000 × 5.72 = 5720 lb. The refrigeration required of the brine in the

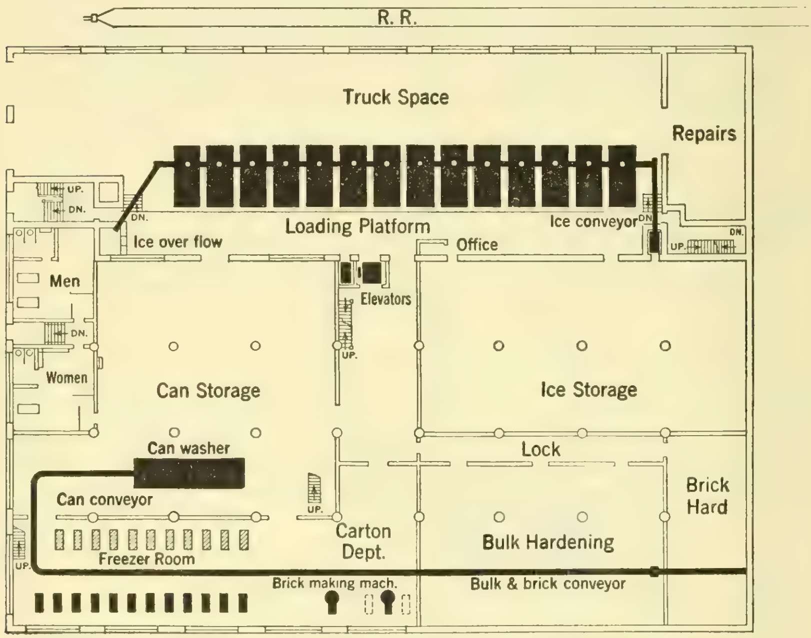


FIG. 355.—Ice Cream Manufacture.

freezers, if the mix enters the freezers at 40 deg. F., would be (80 − 24) × 5720 = 320,300 B.t.u. The brine for the purpose would be at from 0 deg. to 10 deg. F. Adding 10 per cent for leakage and other losses gives a total of 352,300 B.t.u. of refrigeration required of the freezers per 8 hours. The refrigerating work in the hardening room would be (130 − 80) × 5720 = 286,000 B.t.u., or 314,600 B.t.u. with the same allowance for losses. If the cream is packed in 2½ and 5-gal. cylindrical containers, these measure 9 in. diameter by 13 in. high and weigh 11½ lb. for the smaller and 9 in. diameter by 23 in. high and weigh 19 lb. for the larger. A hardening room of about the dimensions shown in Fig. 346 would need to be 10 ft. deep for this amount of cream, and the dimensions of the room would be 11 ft. 2¼ in. × 10 × 9 ft. 6 in. high inside. With 8-in. corkboard all around and 90 degrees outside tem-

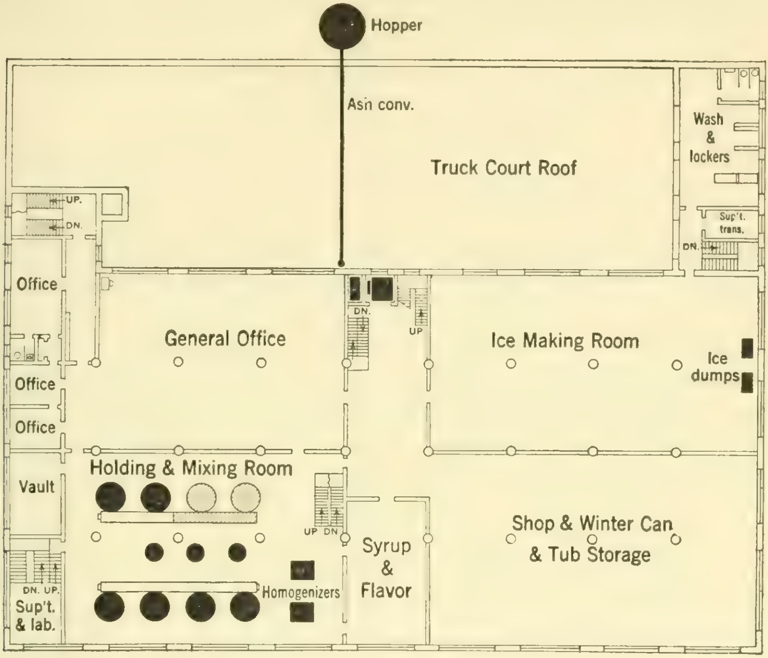


Fig. 356.—Ice Cream Manufacture.

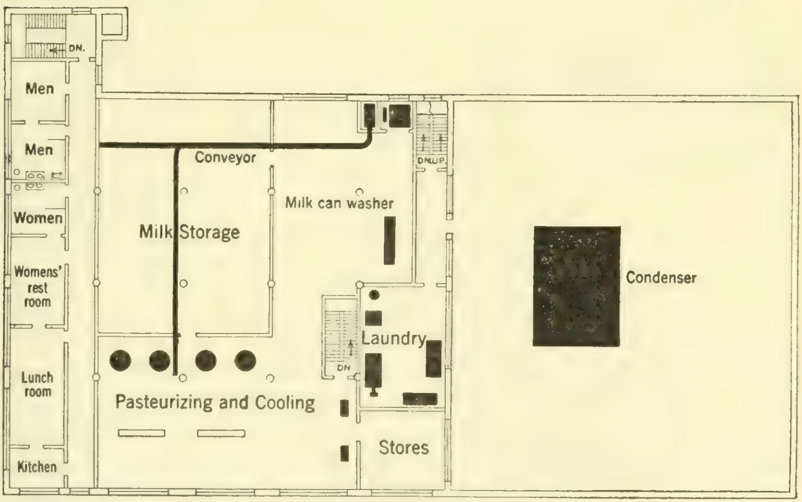


Fig. 357.—Ice Cream Manufacture.

perature of the air and − 10 degrees inside, the heat leakage is, without allowing for the insulating value of the floor:

$$U = \frac{1}{\frac{1}{1.4} + \frac{1}{1.4} + \frac{8.0}{0.296}} = 0.0352.$$

Therefore,

$$Q = 622.9 \times 0.0352 \times [90 - (-10)] = 2200 \text{ B.t.u. per hour.}$$

Such an excessive temperature difference would not be found during the night time except under very unusual conditions. Frequently the insulation will hold the hardening room during the night time without more than 3 to 5 degrees rise of temperature.

Figure 351 gives some details of brine circulation in the creamery, and Figs. 353 to 357 give the general arrangement in a well-designed ice cream factory. The latter has been designed with provision for expansion as well as an economical routing of the material and finished products.

TABLE 114

VANILLA ICE CREAM, OVER RUN 100 PER CENT

(Initial Temperature of Mix 70 deg. F.)

	Before Whipping	After Whipping Refrigeration Shut Off	After 2 Days in Hardening Room	After 7 Days in Hardening Room
Amount of water frozen into ice (per cent)	40.0	20.0	65.0	94.0
B.t.u. of refrigeration per lb. of ice cream absorbed by the mix (total)	92.0	78.0	....	138.0

CHAPTER XVII

HOTEL AND APARTMENT REFRIGERATION

Mechanical refrigeration is now a necessity in the modern hotel to the same extent that the ice cooled box was a few years ago. The larger hotel or club has a large number of special boxes requiring refrigeration (Table 115), located convenient to the workroom or where service or storage is required. These boxes may be quite small, or of fair size, depending on the kind of box. In addition, some hotels make their own ice, calculated from the ratio of 1.6 to 2 tons of refrigeration per ton of ice making, their own ice cream and cool the drinking water supplied to the rooms and dining rooms. The refrigerating load is somewhat indefinite even after the details of the cold storage boxes and other details are known, but an estimate can be made with some degree of precision. For example, the refrigeration for the small boxes can be estimated on the liberal basis of 5.0 B.t.u. per sq. ft. per degree difference of temperature per 24 hours for the heat leakage, with 50 per cent added for the opening of doors, whereas the larger boxes may have a loss due to leakage of only 2.0 B.t.u. and the large cold storage boxes only 1.0 B.t.u. per sq. ft. per degree difference of temperature per 24 hours. These are considered to be very liberal values, in order to take into account the possible poor insulation and the probable poor maintenance of good conditions. As a rule the piping is large by about 25 per cent or more, and less piping will be found to work satisfactorily. For small boxes up to 500 cu. ft. capacity it is usual to use 1-in. pipe, and from 500 to 2000 cu. ft. $1\frac{1}{4}$-in. pipe is usual.

As an example of the use of the table and the chart for the requirements for hotel refrigeration as applied to refrigerator boxes the pantry box will be taken. This is marked B 36, which refers to a box in frequent use to be maintained at a temperature of 36 deg. F. The piping ratio of one such of 500 cu. ft. capacity (Fig. 285) is given as 1.5 cu. ft. per 1 lin. ft. of 1-in. brine pipe, or 334 ft. of pipe will be required. If the outside temperature is taken as 80 deg. F. and the outside area of the box is 390 sq. ft., taking a value of 2.5 for the coefficient of heat transfer of the walls of the box, then the leakage will be

$$Q = 390 \times 2.5 \times (80 - 36) = 42{,}900 \text{ B.t.u. per 24 hours,}$$

to which will be added 50 per cent, or 21,450 B.t.u., per 24 hours for the live load, and the total will be 64,350 B.t.u., or 0.223 tons. The amount of piping taken from the chart, 334 lin. ft. of 1-in. pipe, will give a heat transfer of

$$\frac{334}{2.90} \times 2.0 \times 24 \times (36 - 20) = 88,450 \text{ B.t.u. per 24 hours,}$$
$$= 0.307 \text{ ton.}$$

TABLE 115

HOTEL AND RESTAURANT BOXES

	Rating	Temperature		Rating	Temperature
Baker Shop:			*Kitchen* (Con.):		
Ice cream	A	28	Serving box	B	36
Pantry	B	36 to 38	Serving pantry	B	36
Pastry	B	36	Terrapin kitchen	B	38
			Vegetables	C	38–40
Café and Restaurant:			Sundries	B	36
Lunch box	B	38	Pickle box	C	38
Pantry	B	36	Salad (serving pantry)	C	36
Refrigerating box:					
Nurses, officers,			*In Storage Room:*		
private	B	38	Butter, small	B	36
			Cheese, serving box	B	36
Kitchen:			Cigars	C	36
Corned beef	C	36	Cream	B	36
Pastry counter	B	36	Eggs, serving box	B	31
Dried fish	C	36	Fruit	B	38
Fresh fish	A	25 to 30	Milk	B	36
Freezer	A	15 to 18	Pantry	B	38
Garde manger	B	36 to 38	Receiving room	B	36
Meat-cooler	C	38	Storage box	B	36
Meat cut	B	35	Storage box	C	36
Order box chef	B	36			
Oysters, live	C	43 to 45	*Sundries:*		
Oysters, in tubs	B	36	Carafe	A	28
Pantry	B	36	Ice storage	A	26
Poultry and Game	A	28	Swill	B	36

Taking the curves for the cubic feet of room space per ton of refrigeration (Fig. 285), the value given for a 500-cu. ft. capacity box, and 36 deg. F., is 1400 cu. ft. per ton of refrigeration, the tonnage becomes

$\frac{500}{1400} = 0.36$. This value, 0.36 ton, is larger than either of the calculated values of tonnage in the preceding paragraph, and it indicates that the tables and curves are calculated liberally and are conservative, so as to allow for all commercial factors of construction and operation. It is usually agreed that direct expansion piping is likely to give less heat transfer than brine piping will, because the ammonia (or carbon dioxide) in evaporating partly fills the inside of the pipe with gas, thereby making the surface inefficient for heat transfer.

The total requirements for hotel refrigeration can be obtained only by a careful computation of the separate cold storage boxes and their operating conditions and all the other uses of refrigeration. Proper allowance must be made for pipe line and other losses.

Apartment Refrigeration.—All modern apartments of any size are provided with mechanical refrigeration, and this is almost always by means of brine rather than by direct expansion because of better control and because of the greater danger from ammonia were that to be used. It should be noted, however, that carbon dioxide has been popular in this application because of the lack of danger should it happen that a heavy leak should occur. The usual dimensions for apartment boxes vary from 18 in. by 20 in. by 30 in. to 22 in. by 44 in. by 60 in., with 6 cu. ft. and 30 cu. ft. displacement respectively, but the average displacement is 15 cu. ft., corresponding to a box 18 in. by 30 in. by 48 in. The external area of these boxes varies from 20 to 65, with an average of about 40 sq. ft.

The Refrigeration Required.—In calculating the total refrigeration required some idea of the details of the installation is necessary. Without question a better grade of refrigerator now is used, i.e., one constructed with 1 in. to 2 in. corkboard. Such tests as are available indicate a leakage loss through the construction of from 0.15 to 0.18, with the 2-in. and the 1-in. corkboard boxes respectively, B.t.u. per sq. ft. per degree difference per hour. The cooling surfaces installed must overcome the heat leakages and the live load due to the opening of the doors and the insertion of the hot and warm commodities. Such surfaces are calculated on the basis of 1.7 B.t.u. as the value of k. If pipe coils are used, the $\frac{3}{4}$-in. pipe is the more usual, but cast iron sections or cast iron "brine boxes" are becoming popular, as well as welded boxes made up of sheet iron. Most of these brine boxes (snow mounds) are designed to hold small ice cube pans so as to make small amounts of ice (Fig. 358). The brine temperature carried is usually 15 deg. F. when ice is made, and from 20 to 25 degrees in other cases. The temperature in the box is usually, when brine refrigerated, kept at about 40 degrees.

Piping.—The brine piping for apartment refrigeration is usually arranged as shown in Figs. 361, 169 and 362, so as to be balanced. The

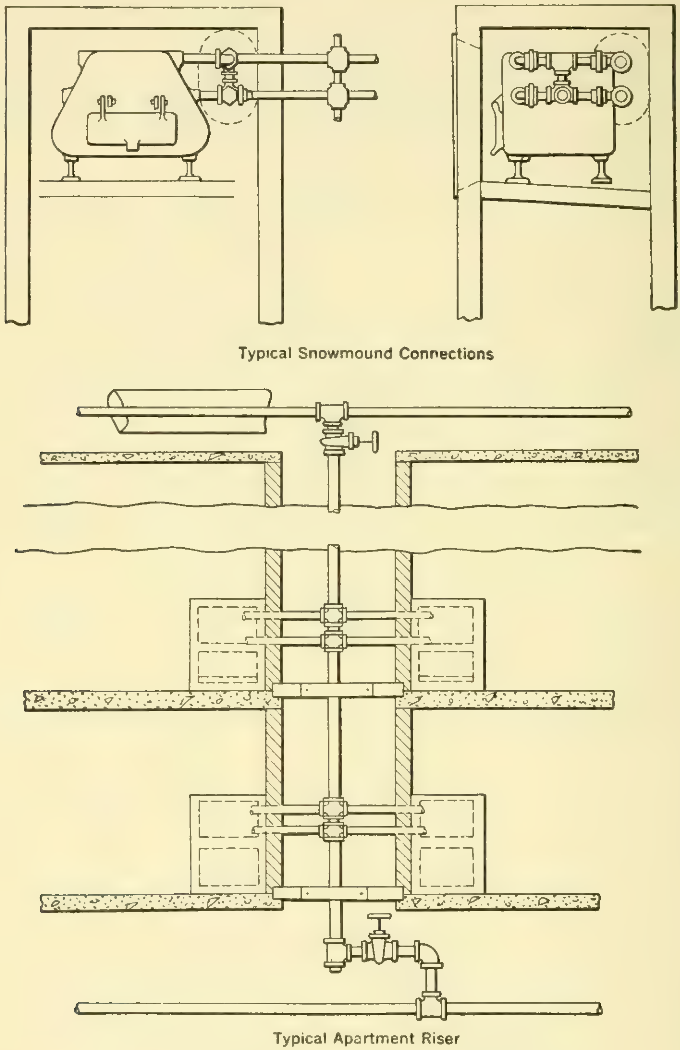
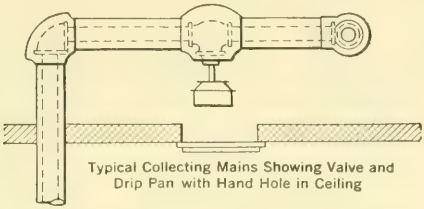


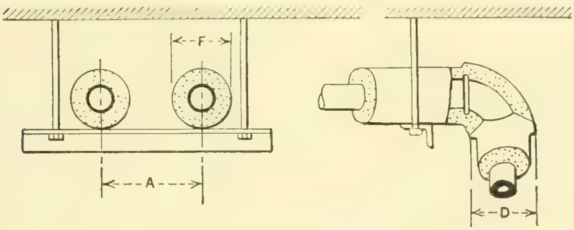
FIG. 358.—Typical Apartment Piping.

shell and coil, the shell and tube and the double pipe brine cooler may be used, with or without brine storage tanks. If no storage tank is used it is necessary to operate the compressor whenever refrigeration is

required as the amount of the brine in the system is small. The double-riser system allows brine of practically the same temperature to enter each box, and the system is not limited to any maximum number of boxes per riser, but the first cost is much greater because of this greater amount of piping. The single riser is popular, especially in apartments,



Typical Collecting Mains Showing Valve and Drip Pan with Hand Hole in Ceiling



Minimum Dimensions						
Size of pipe		A		D		F
½"		6¼"		4½"		4¼"
¾"		6¾"		6"		4¾"
1"		7⅜"		5⅞"		5 5/16"
1¼"		8⅜"		6⅝"		6 3/16"
1½"		9"		7¼"		6⅞"
2"		9½"		7½"		7¼"
2½"		11"		8⅛"		7⅞"
3"		12"		9⅛"		8⅞"
3½"		13"		10"		9⅝"

Typical Hanger and Dimensions to Allow Application of Cork Covering

Fig. 359.—Typical Hanger and Details.

as about half the amount of piping and pipe covering is used, and one-half of the cost of erection, space occupied and the amount of joints with their possible leaks will occur. But with the single-pipe system (Fig. 362) all the brine is required to pass through each box, unless it is by-passed, and the last box receives the highest temperature brine. The single riser is usually limited to 12 to 14 boxes per riser. A brine velocity of not more than 5 ft. per second should be used, and not more

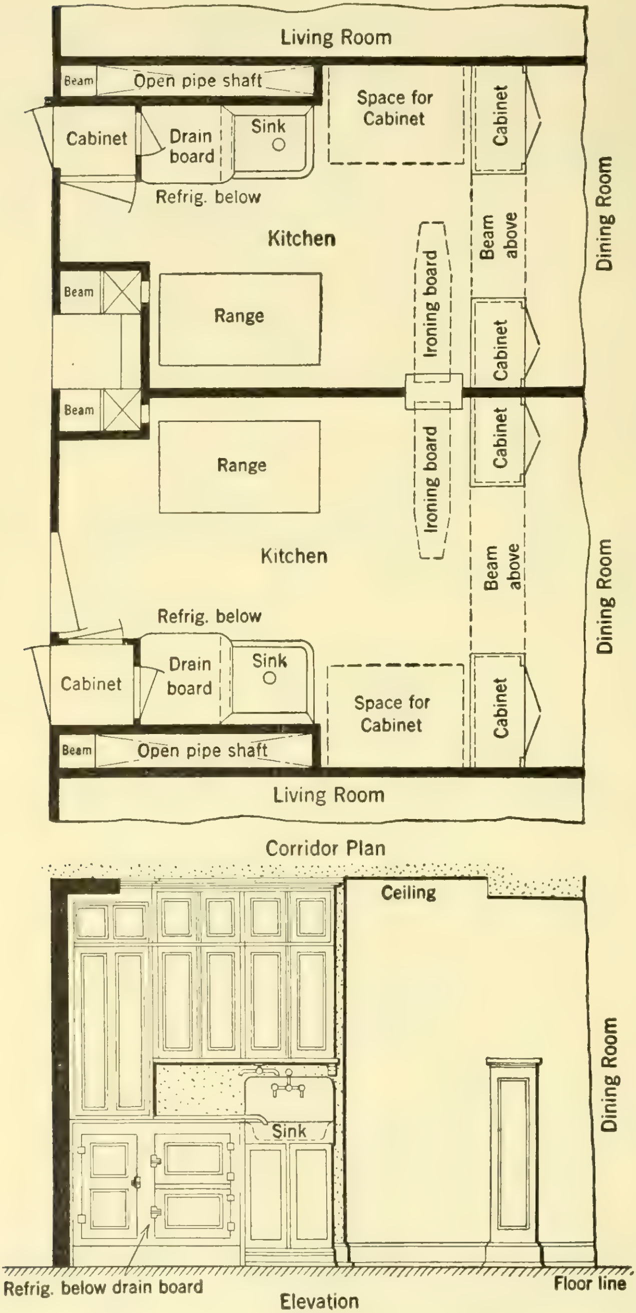


FIG. 360.—Typical Arrangement of Apartment.

than 3.0 deg. F. rise of temperature of the brine should be permitted during the circuit through the boxes.

The pumping head varies with each installation. With the balanced tank design the head is due to the friction head and the velocity head only. This may be calculated as in the problem on brine piping, Chapter VIII. Allowance for the fittings should always be made in calculating

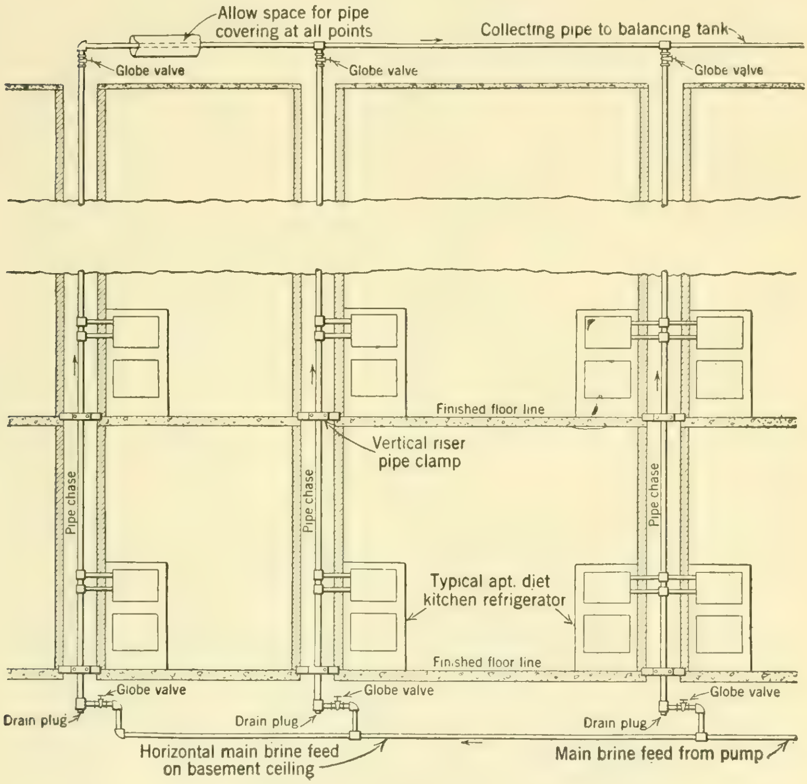


Fig. 361.—Apartment Piping.

the total head on the pump. The pressure on the brine tank or the piping at the pump is the " total head " plus the static head due to the elevation of the balance tank above the pump. Figures 350 to 361 give typical piping arrangements and typical apartment arrangment of boxes.

The Drinking Water System.—Factories and particularly office buildings and hotels are making use of the mechanically cooled drinking

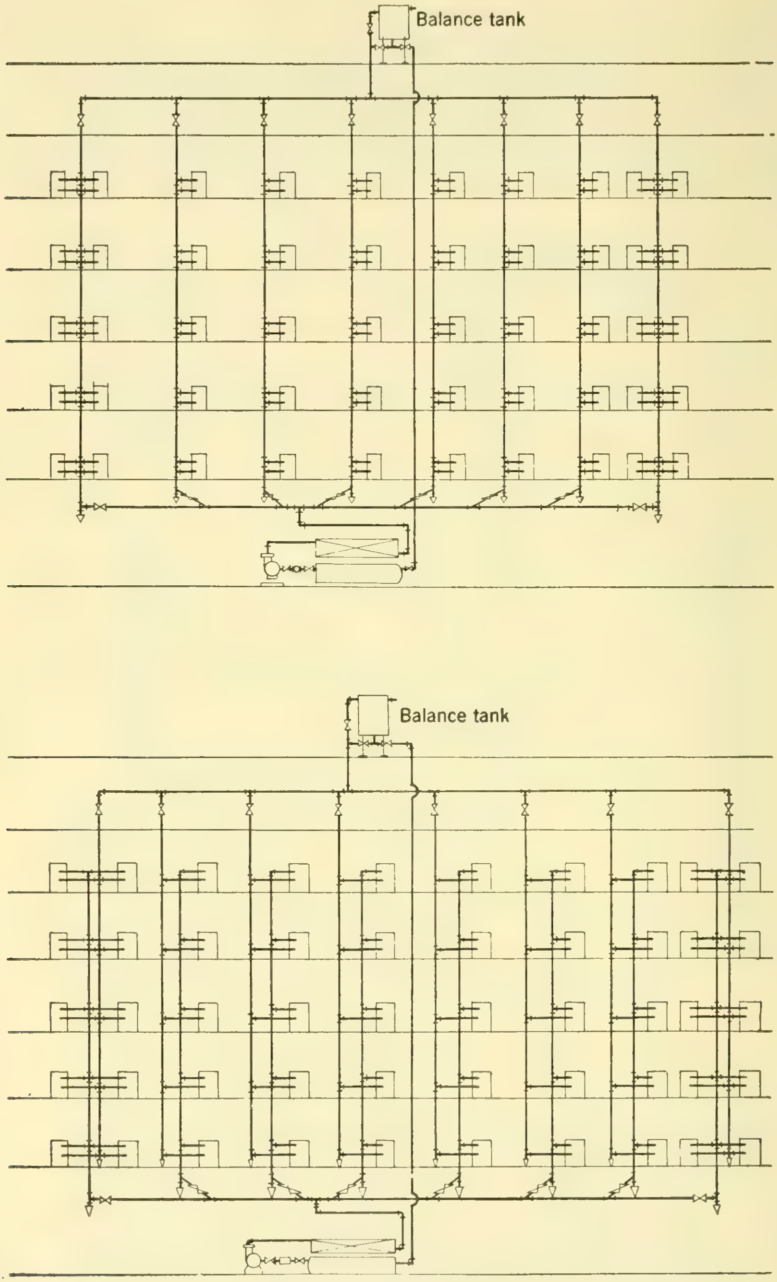


FIG. 362.—Brine Piping.

water system, as the use of the ice cooler is neither convenient, economical, nor hygienic. With drinking water systems the water may be

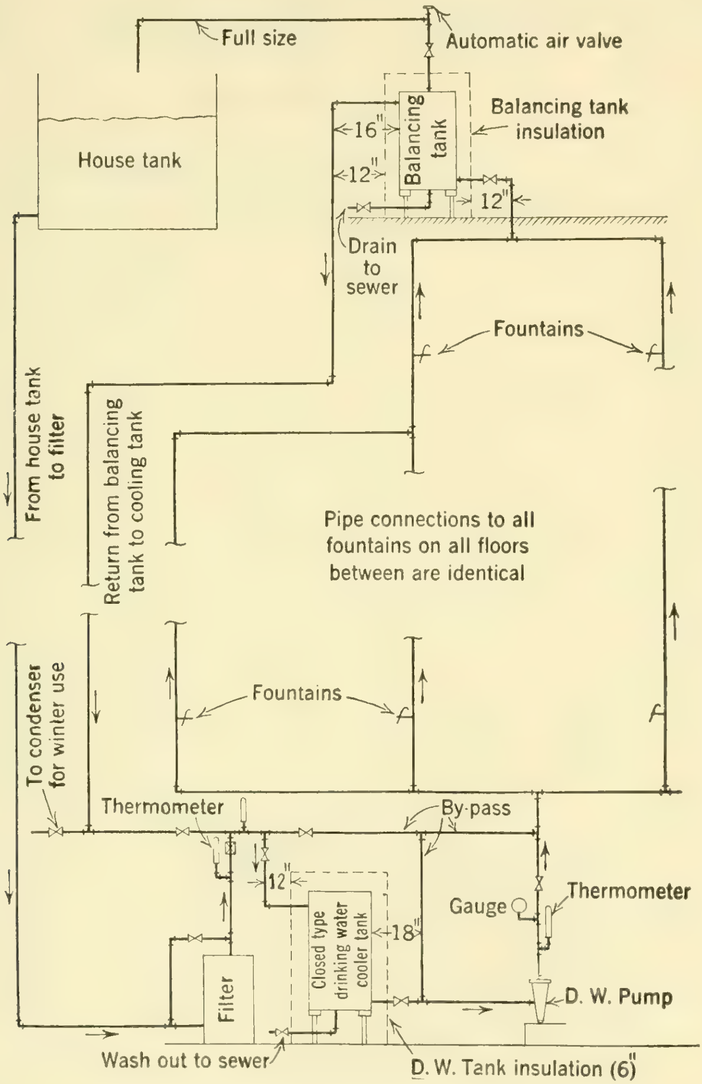


FIG. 363.—Drinking Water System.

filtered, is cooled to the desired temperature and, in particular, may be designed with convenient fountains for the use of the workmen or the office force in the different parts of the factory or office.

Such a refrigerating system consists of a water-cooling tank of one design or another, the refrigerating unit, a circulating pump to keep the water in constant circulation throughout the system, a balance tank and, finally, a make-up valve and float so as to replenish the water

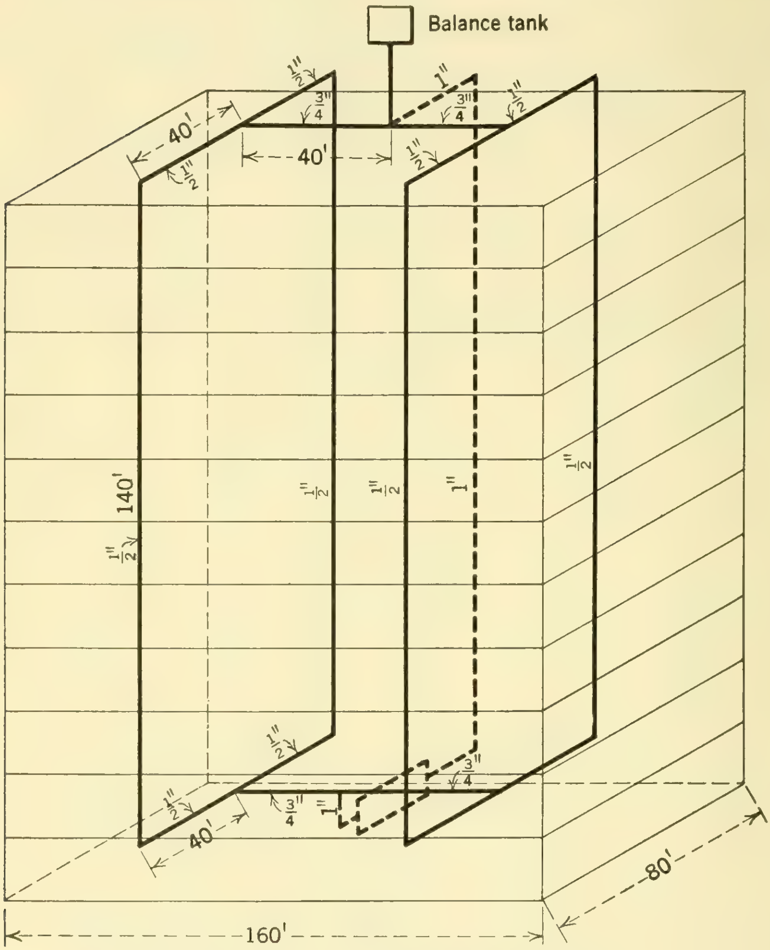


FIG. 364.—Problem in Drinking Water Systems.

supply as fast as it is used, sometimes in connection with the balance tank. Figure 363 shows a typical installation.

The water circuit must be complete at all times. By this is meant that from four to five times the water consumed, and wasted, must be

circulated in the pipes. This circulating water is required so as to have a positive pressure always at the outlets of all the fountains, and also so as to absorb and carry away the heat leakage through the pipe insulation. As the water temperature may be assumed to be 50 deg. F., and the outside air temperature 95 degrees, the heat leakage develops into large proportions at times.

The function of the refrigerating machine then is to cool the make-up water to, say, 50 degrees and to overcome the effect of heat leakage. The latter naturally depends on the kind of insulation used and the length and the size of the pipe in the water circuit. It has been found advantageous to use $\frac{3}{4}$ and 1-in. full weight pipe for the circuits as a rule, and to limit the circuits to 2000 ft., although 5000 ft. circuits have been successful. Too long a water circuit will represent an excessive pumping cost, and in addition to the increased power this work of pumping invariably will degenerate into a heating effect which in turn will have to be corrected by refrigeration at the rate of $\frac{1}{5}$ ton refrigeration per one horse power of the circulating pump. For the same reason the velocity of the water in the pipe should be limited to not more than 200 ft. per minute.

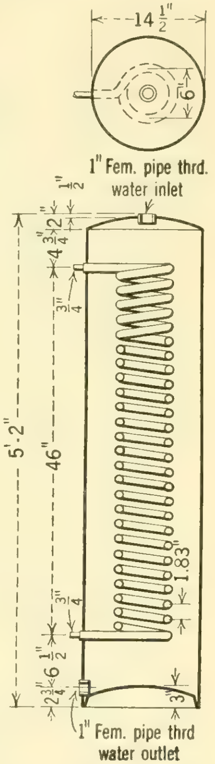


FIG. 365.—Drinking Water Cooler.

The amount of water *used and wasted* is a difficult variable to account for. It naturally is dependent on the severity of the labor of the workmen. Hotels and office buildings having an outlet in each room should be designed on the basis of *one gallon* per person per day. Figure 368 shows the amount of water consumed in one of the St. Louis hotels. This plant was laid out to take care of a demand of 1000 gal. per hour or 1.5 gal per faucet per hour. In all likelihood one-sixth of this amount would have been sufficient, or on the basis of $\frac{1}{4}$ gal. per faucet per hour. So it would seem that the rule of one gallon per person per day of eight hours is very liberal, although it will always be safe. Stores and office buildings providing outlets in isolated places in aisles or corridors may have these in use continuously, and they should be figured on the basis of a *half pint* per minute per fountain. In factories it is good practice to use *one-half gallon* per person per hour, and in offices *one-quarter gallon* per person per hour. These figures are much higher than those used in

the Federal Buildings,[1] which are designed with the uniform rule of *one pint* of water per person per hour irrespective of the kind of service the building may be called upon for.

The number of fountains are also variable, depending as this does on the kind of installation. The attempt is made in their location to

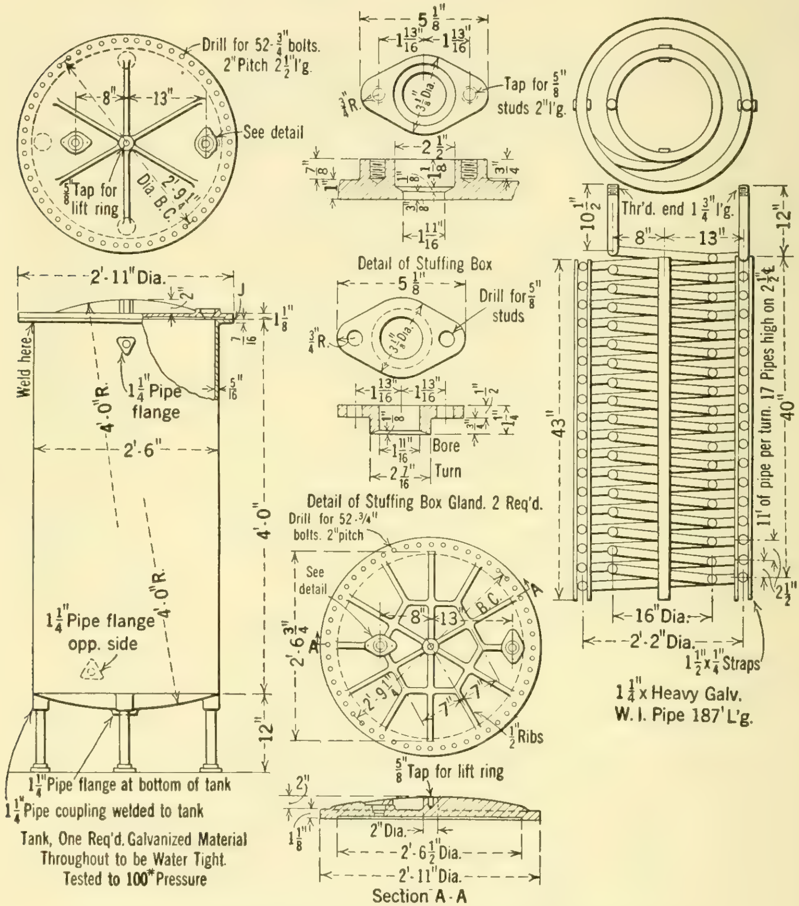


Fig. 366.—Drinking Water Cooled by Means of Carbon Dioxide.

occasion as little difficulty as possible for each workman, and to work on the assumption that the first cost of extra fountains is preferable to the cost of the lost time of the workman in walking to a distant fountain. In consequence, good practice will allow one fountain per 100 operators

[1] Nelson S. Thompson, Supervising Architects Office, Treasury Department.

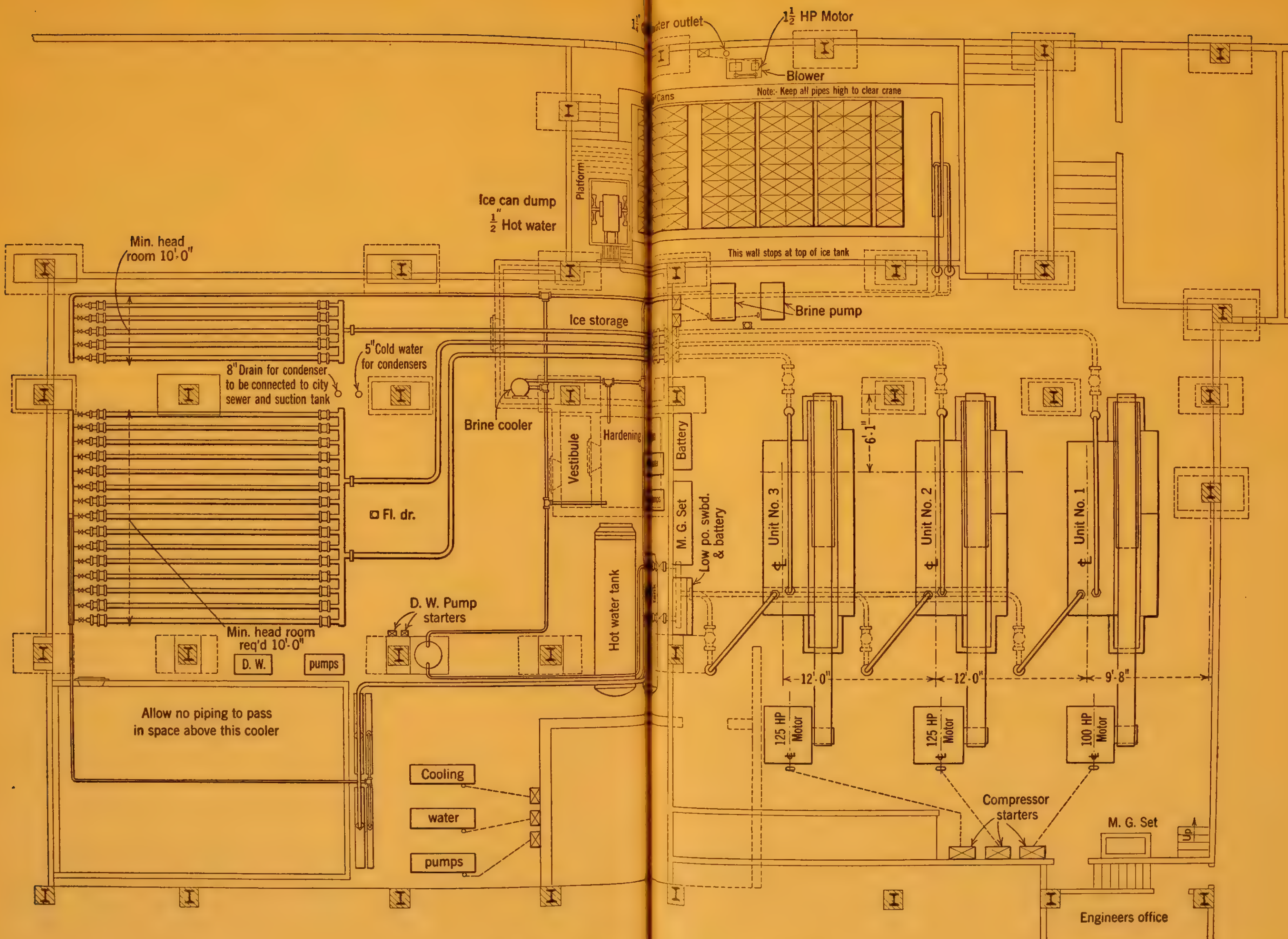


FIG. 367.—Hotel Refrigeration, using Carbon Dioxide.

To face page 579.

or per 50 workmen, depending on the details of the job and the congestion of the shop or office.

The kind of insulation used is also very important from the viewpoint of the heat leakage, and Table 116 can be used advantageously for ice water conditions, the corkboard and the lithboard being $1\frac{1}{2}$ in. thick and the hair felt being erected in two layers. The heat transfer values are for *one foot* of length per *one degree* difference in temperature *per hour.*

TABLE 116

Coefficient of Heat Transfer, per Linear Foot

	Union Lith	Hair Felt (Two Layers)	Corkboard
$\frac{1}{2}$-in. pipe	0.157	0.086	0.160
$\frac{3}{4}$-in. pipe	0.170	0.095	0.167
1 -in. pipe	0.180	0.107	0.178
$1\frac{1}{4}$-in. pipe	0.213	0.124	0.199
$1\frac{1}{2}$-in. pipe	0.217	0.132	0.220
2 -in. pipe	0.269	0.149	0.245
$2\frac{1}{2}$-in. pipe	0.272	0.167	0.291
3 -in. pipe	0.294	0.192	0.304
$3\frac{1}{2}$-in. pipe	0.355	0.211	0.330
4 -in. pipe	0.367	0.228	0.345
$4\frac{1}{2}$-in. pipe	0.376	0.247	0.385
5 -in. pipe	0.466	0.268	0.410
6 -in. pipe	0.558	0.307	0.437

Problem.—The following problem illustrates the manner of calculation of the size of a drinking water system. An office building 160 by 80 ft. and 10 stories high is to have a drinking water system with the water to be delivered at a temperature of 50 deg. F. The floors are 13 ft. high, the make-up water is at a temperature of 80 degrees and 3.0 gal. are to be provided per person per day of 8 hours.

The floor area is 12,800 sq. ft. and, with an allowance of one person per 70 sq. ft.[2] of floor area, there will be $\frac{12,800}{70} = 183$ persons per floor. Using one fountain per 50 persons there will be required 4 fountains per floor and 40 fountains in all. The piping will be arranged as shown in the figure, which has four circuits in parallel. Assume that the risers will be of $\frac{1}{2}$-in. pipe using 225 ft. of $\frac{1}{2}$-in. pipe, 80 ft. of $\frac{3}{4}$-in. pipe, and 220 ft. of 1-in. pipe in each of the circuits, except that the 1-in. pipe is the common return. The heat absorbed using an average temperature of the water of 47.5 degrees, 95 degrees outside temperature, and corkboard covering, is as follows:

[2] The factory allows one person per 100 sq. ft.

½-in. line. $Q = 225 \times 0.160 \times (95 - 47.5) = 1710$ B.t.u. per hour (per riser)

$= 6840$ B.t.u. per hour.

¾-in. line. $Q = 160 \times 0.167 \times (95 - 47.5) = 1270$ B.t.u.

1-*in. line.* $Q = 220 \times 0.178 \times (95 - 47.5) = 1860$ B.t.u.

Total $= 9970$ B.t.u. per hour.

The amount of water required to absorb this heat, assuming a rise of temperature of the water of $4\frac{1}{2}$ degrees,

$$\frac{9970}{4.5} = 2216 \text{ lb. per hour}$$

$$= 4.43 \text{ gal. per min. total}$$
$$= 1.11 \text{ gal. per min. per circuit.}$$

Make-up water. $3.0 \times 10 \times 183 = 5490$ gal. per 8 hr.
$= 686$ gal. per hr. $= 11.45$ gal. per min.
$= 2.86$ gal. per min. per riser.

Total water required, neglecting the effect of friction heating $= 15.88$ gal. per min.

Refrigeration required.

$$8.33 \times 60 \times 11.45 \times (80 - 45) + 9970 = 210{,}400 \text{ B.t.u.} = 17.6 \text{ tons.}$$

The friction developed in the water circuit is (Fig. 180):

½-in. line (3.97 gal. per min.) = 2.6 in. of water per ft.
¾-in. line (7.94 gal. per min.) = 2.3 in. of water per ft.
1-in. line (15.88 gal. per min.) = 2.3 in. of water per ft.

Total friction $= (2.6 \times 225) + (2.3 \times 80) + (2.3 \times 220) = 1275$ in. of water
$= 46.0$ lb. per sq. in.

The heating effect of the friction head is

$$15.88 \times 8.33 \times 46.0 \times 2.31 = 14{,}050 \text{ ft.-lb. per min.}$$

$$\frac{14{,}050}{33{,}000} \times 42.4 = 18.1 \text{ B.t.u. per min.}$$

The size of the motor will be $0.425 \div 0.5 = 0.850 = 1.00$ hp.

Figure 365 gives details of a small water cooler and Fig. 366 gives the construction of a typical water cooler using carbon dioxide as a refrigerant. The carbon dioxide under these circumstances is always *placed in the pipes.* Figure 367 gives a representative layout for carbonic refrigeration in a hotel. Figure 368 shows graphically the demand on a drinking water system in the Statler Hotel, St. Louis, for a single day and for a five-day period. This system was designed for a maximum demand of 1000 gal. per hour.

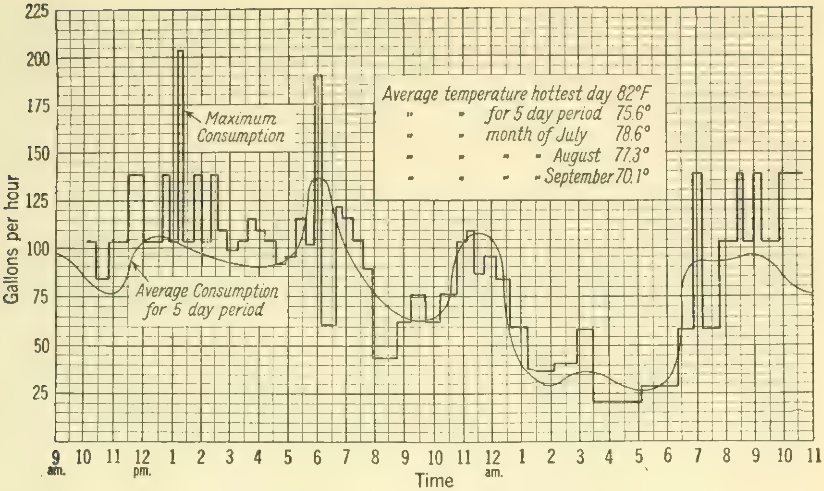


Fig. 368.—Load Curve for Hotel Drinking Water.

TABLE 117

Piping Ratio for Small Refrigerators—Number of Cubic Feet of Space Cooled by One Running Foot of Brine Coil

For Refrigeration up to 200 Cubic Feet		For Refrigeration up to 500 Cubic Feet		For Refrigeration up to 1000 Cubic Feet		For Refrigeration up to 2000 Cubic Feet	
Temperature, degrees F.	Cubic feet	Temperature, degrees F.	Cubic feet	Temperature, degrees F.	Cubic feet	Temperature, degrees F.	Cubic feet
40	3.0	40	4.0	40	8.0	40	10.0
36	2.0	36	3.0	36	6.25	36	8.0
32	1.75	32	2.5	32	5.0	32	6.5
20	1.25	20	1.75	20	3.75	20	5.25
10	.7	10	1.25	10	2.50	10	4.0
5	.5	5	.75	5	1.25	5	2.25
0	.25	0	.35	0	.625	0	1.25

Sizes of Brine Mains for Small Refrigerators

$\frac{1}{2}$-in. Feed and return for $1\frac{1}{4}$-in. Coils up to 200 lineal feet.
$\frac{3}{4}$-in. Feed and return for $1\frac{1}{4}$-in. Coils up to 400 lineal feet.
1 -in. Feed and return for $1\frac{1}{4}$-in. Coils up to 800 lineal feet.
$1\frac{1}{4}$-in. Feed and return for $1\frac{1}{4}$-in. Coils up to 1500 lineal feet.

CHAPTER XVIII

REFRIGERATION IN THE CHEMICAL INDUSTRIES

There is no question but that mechanical refrigeration will play an increasingly important part in the development of processes in the chemical industry. Mechanical cooling may be used to hasten the process, but at times it is absolutely necessary not only to condition the air or to secure moderately cold temperatures, but to obtain very low temperatures. As a rule the actual cooling problem is relatively simple, as it is usually simply the cooling of a liquid, or the cooling and freezing of a liquid, and the constants involved in the calculation of this process can be secured only by experiment. A large number of the chemical processes are secret, but a few of the more generally known applications will be given in this chapter with the understanding that other processes are similar in their method of solution.

Low Temperatures.—In America low temperatures have been secured by the use, first, of the absorption machine and, more lately, by the use of the stage compressor. The Eastman Kodak Company, which uses in their process work a − 20 deg. F. brine, has three 700-ton two-stage ammonia compressors. In Great Britain low temperature refrigeration has been obtained by means of the carbonic compressor which has been developed to a much higher degree than it has in the United States. For − 60 deg. F. refrigeration the two-stage carbonic compressor, built by the Liverpool Refrigerating Company, has had outstanding success. The pressure in the evaporator for − 60 deg. F. is about 100 lb. per sq. in. with carbon dioxide, whereas with ammonia it would be only 5.5 lb. absolute.

Cooling Corrosive Substances.—Corrosive substances can be cooled best by the use of brine. This brine piping can be designed for slight unit pressures and the piping can be made of lead, porcelain covered, or even of glass.

Removal of Salts from Solution.—Certain salts may be extracted best by means of crystallization and precipitation incidental to a lowering of the temperature. For example, a saturated solution of sodium nitrate, $NaNO_3$, at 90 deg. C., will hold in solution 162 grams of salt in 100 cc. of water but at 20 deg. C. the amount will be reduced to 85 grams and

the difference (77 grams) must be precipitated out of solution. The following problem is a good example of this sort of application of refrigeration.

Problem.—10,000 lb. of water saturated with potassium chlorate, $KClO_3$, is to be cooled from 68 deg. F. to 14 deg. F. per hour. Find the refrigeration and the piping required for the heat transfer.

One arrangement of the apparatus required is shown in (Fig. 369). This apparatus consists of two cylindrical tanks with conical bottoms, the latter fitted with cocks of ample size for quick operation. Referring to the figure it will be seen that the strong solution enters the left-hand tank at 68 deg. F. and passes downward, being cooled on the way, with precipitation of some potassium chlorate

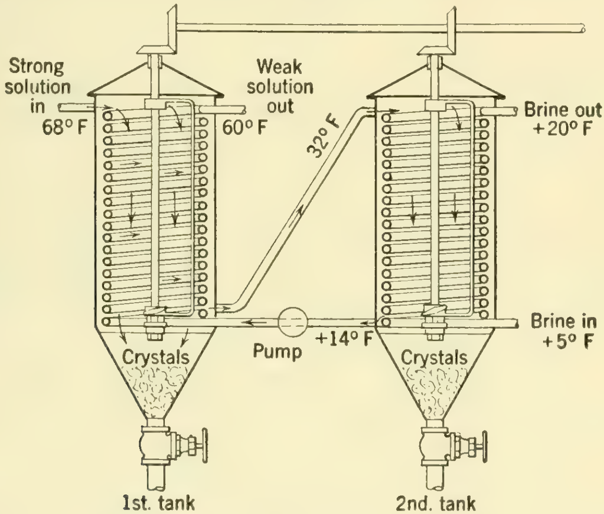


Fig. 369.—Removal of Salts from Solution.

crystals, and finally passes over into the right-hand tank by means of the difference in liquid level or by means of a pump. In the second tank is arranged a helical coil for brine cooling, the brine entering at 5 deg. and leaving at 20 deg. F. The solution is thereby lowered in temperature to 14 degrees and leaves the tank at this temperature to be pumped through a helical coil in the first tank where it leaves at 60 deg. F. The first tank (the one on the left) is therefore simply a heat exchanger, intended to give economy of action by the utilization of all the refrigeration existing in the liquid after the process that it is practical to attempt to use.

As the crystals will tend to form on the cooling coils it is necessary to provide some means of continually scraping them, and this is done by the wire brushes attached to the vertical shafts, as shown in the figure. The crystals settle into the conical part of the tank and are drawn off into other tanks, the upper part of which is separated from the lower by means of wire screen so that what liquid passes along with the solid potassium chlorate will pass through the screen.

To precipitate the potassium chlorate there will be required 146.7 B.t.u. of refrigeration per pound of salt formed. Referring to the chart (Fig. 370), $(7.4 - 2.2) =$

5.2 grams of salt is precipitated per 100 cc. of water, or 5.2 per cent of the weight of the water is the weight of the salt removed from solution, therefore 520 lb. of salt crystals will be produced per hour. During the process in the first tank the weak solution will be heated from 14 to 60 deg. F., and the heat absorbed, taking the specific heat of the solution at 0.85 will be:

$$Q = 10{,}220 \times 0.85 \times (60 - 14) = 400{,}000 \text{ B.t.u.},$$

and the strong solution will lose an equal amount of heat as heat of the liquid and as heat of the solution. By trial and error it works out that, by allowing for losses through the insulation the temperature of the strong solution leaving the first tank will be 30 degrees, and that the strong solution will be reduced to a concentration of 3.3 grams per 100 cc. of water whereby 410 lb. of salt will be precipitated and the remaining 110 lb. of chlorate will be deposited in the tank with the brine coil.

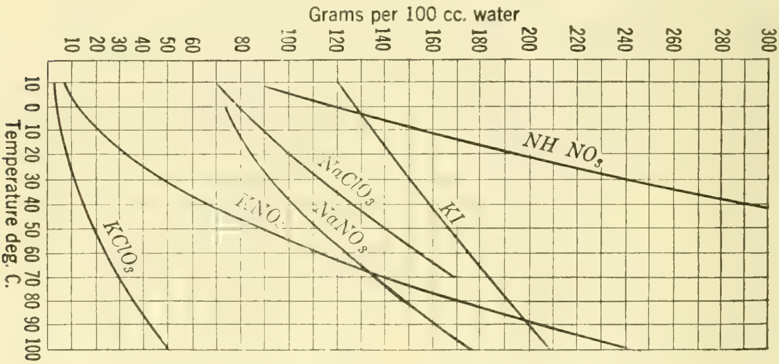


FIG. 370.—Solubility of Salts at Various Temperatures.

The load on the brine coil will be (taking the heat of precipitation of the salt solution as 146.7 B.t.u.),

$$Q = (10{,}270 \times 0.85 \times 30 - 14) + (110 \times 146.7) = 155{,}800 \text{ B.t.u.}$$

$$= 12.99 \text{ tons of refrigeration.}$$

To this 13.0 tons of refrigeration must be added 5 per cent for losses through the insulation, which makes the total load on the refrigerating machine 13.64 tons. The compressor displacement, for 15 lb. suction and 160 condenser pressure, will be $13.64 \times 4.7 = 64.1$ cu. ft. per minute, and the power required to drive the compressor will be $1.42 \times 13.64 = 19.37$ hp., but a larger motor will be required for starting and as a factor of safety. For this size of compressor the best design is the single-acting vertical enclosed type of machine which will operate at about 200 r.p.m. Using these values, the size of compressor works out to be 7.1 by 7.1, so that the 7 by 7 or the $7\frac{1}{2}$ by $7\frac{1}{2}$ standard sizes would be used, making the proper adjustment of speed to suit.

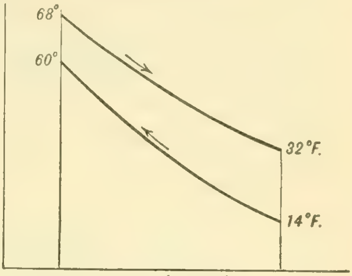
The amount of pipe surface for the coils in the tanks may be considered from the following viewpoint. The inside of the coils will be kept free of crystals because of

the action of the rotating brushes, and the propeller will keep the liquid agitated. A value of 50 to 75 for the coefficient of heat transfer is justified if all the surface is effective, but as half is coated with salt it will be safer to use a value of 40. The mean temperature difference in the first tank (the logarithmic mean) is 12.35 (Fig. 163 and 371), and the surface becomes:

$$400{,}000 = A \times 40 \times 12.35$$

$$A = 810 \text{ sq. ft.} = 1295 \text{ ft. of 2-in. pipe.}$$

This surface could be provided in a coil 8 ft. in diameter with 52 turns. If the pipe coil has a pitch of 3 in. the coil would be 13 ft. high, with $\frac{5}{8}$ in. between each pipe. In a similar manner the pipe coil required for the second tank would be 410 sq. ft., or 656 lin. ft. of 2-in. pipe. The two tanks should be covered with 4 in. of cork or the equivalent.



$$t_m = \frac{(32-14)-(68-60)}{\log_e \frac{32-14}{68-60}} = 12.35°\text{F.}$$

Fig. 371.—Problem. Method of Securing the Mean Temperature Difference.

Oil Refining.—The oil refineries manufacturing a low "cold test" oil use a procedure almost similar to the process just described except for the details of the plant. The problem again involves the cooling of a liquid and the congealing of a part of this liquid.

The Wax Process.—In the oil refining process, in order to get the required "cold" test for certain grades of lubricating oils it is necessary to chill the paraffine wax distillate and separate the resulting congealed paraffine from the oil by pumping through a filter press. The problem involves the cooling of brine, as direct expansion is seldom used, and a suitable device for chilling the oil. As the wax congeals, some scraping attachment must be included in order that as the solid material collects on the pipe surface it may be scraped away and pushed along in the direction of flow of the chilled oil. The usual design is to use a double pipe cooler with a screw conveyor in the inner pipe (Fig. 372) so constructed as to give a positive flow of the oil and wax and an efficient clearing of the oil surface at all times. It is very necessary to have a clean surface of the piping if results of value are to be obtained.

The amount of wax and the physical constants in distillate vary with the different fields, but the average is taken as 0.5 for the specific heat of the oil, and 0.87 for the specific gravity. The latent heat of fusion of the paraffine is 125 B.t.u. per lb. and it begins to separate out at 60 deg. F. As an illustrative example, 1000 gal. of distillate per hour are to be cooled from 80 to 20 deg. F. and 10 per cent of wax by weight will be removed. Assume the specific heat of the solid wax as 0.6. Find the size of the compressor required, allowing 10 per cent for losses by heat

leakage, etc., and find the size of the double pipe cooler required for cooling the oil.

The weight of oil per hour is $1000 \times 8.33 \times 0.87 = 7{,}250$ lb.

		B.t.u.
a. To cool the oil to 60 deg. F.	$0.5 \times 7250 \times (80 - 60) =$	72,500
b. To freeze the paraffine	$0.1 \times 7250 \times 125 =$	90,600
c. To cool the oil	$0.9 \times 0.5 \times 7250 \times (60 - 20) =$	130,500
d. To cool the wax	$0.1 \times 0.6 \times 7250 \times (60 - 20) =$	17,400
	Total	= 311,000
	Add 10 per cent for safety	= 31,100
	Total[1]	= 342,100
		= 28.5 tons.

The resulting clarified oil, being at 20 degrees, could be used to decrease the refrigerating load by pumping it back to a counterflow double pipe cooler (and permit it to warm up to 40 deg. F. or more), through which the distillate at 80 degrees is passed. In fact the oil cooler is designed at times using the two uppermost pipes as such an "exchanger." If such a device were used, the refrigeration saved could be expected to be:

$$0.9 \times 7250 \times (40 - 20) \times 0.5 = 65{,}250 \text{ B.t.u.},$$

and the tonnage becomes

$$\frac{1.1 \times 245{,}750}{12{,}000} = 22.5 \text{ tons of refrigeration.}$$

But the oil cooler would be less efficient in heat transfer because of having oil on both sides of the inner pipe and also because of the lesser temperature difference when using oil and brine, as compared with brine throughout the cooler, for the two uppermost pipes. As a rule zero degree brine is used in these coolers, and for some time the absorption machine has been the favorite means of securing refrigeration because exhaust steam is always available in refineries and the absorption machine can use this steam directly in the still to generate the high pressure ammonia gas required for liquefaction in the condenser with ordinary condenser water.

The distillate chilling machine, Fig. 372, is made of 4-in. and 2-in. pipes as shown, the brine being in the outer and the distillate in the inner of the pipes. The amount of the cooling surface is more or less empirical as, for example, one refinery uses 400 lin. ft. of cooling pipe surface to

[1] For general design purposes it is usual to follow standard practice of about 7 to 8 barrels of wax distillate per hour per 1 ton of refrigeration.

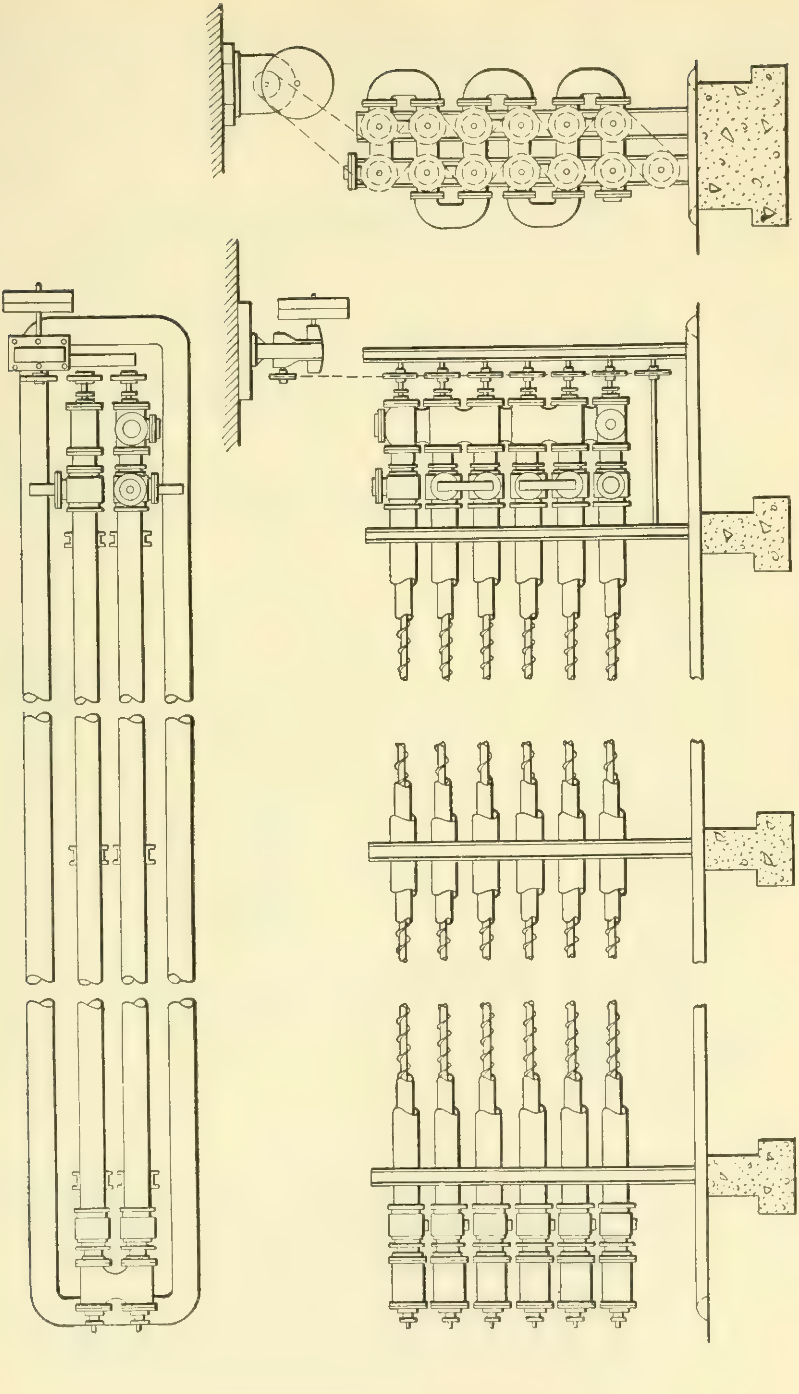


FIG. 372.—The Wax Chilling Machine.

chill 20 gal. of distillate per minute from 80 to 20 degrees using zero degree brine. In this problem the mean temperature difference is

$$t_m = \frac{(80 - 5) - (20 - 0)}{\log_e \dfrac{80 - 5}{20 - 0}} = 41.6 \text{ degrees}$$

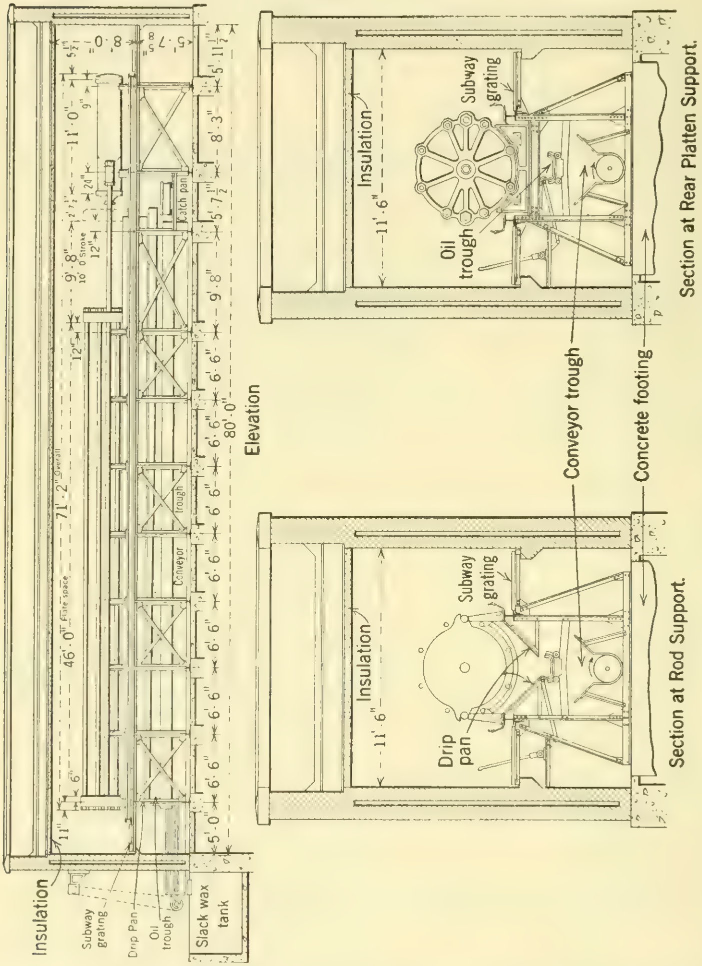


FIG. 373.—Filter Press for Wax Separation.

with a rise of 5 degrees in the temperature of the brine from 0 to 5 deg. F. Using a value of k of 25, the required surface becomes

$$311{,}000 = A \times 25 \times 41.6.$$
$$A = 298 \text{ sq. ft.}$$
$$= 298 \times 1.6 = 478 \text{ linear feet of 2-in. pipe.}$$

The value of 25 for k is possible only when the surface is kept carefully scraped of the wax, which means that the screw conveyor must make a good fit in the inner pipe. Figures 372 and 373 give details of the process. Table 118 contains the constants for the more general sub-

TABLE 118

THERMAL PROPERTIES OF VARIOUS SUBSTANCES

Substance	Temperature, Deg. F.	Specific Gravity	Specific Heat	Latent Heat, B.t.u. per Lb.	Temperature, Deg. F.		
					Freezing	Melting	Boiling
Alcohol, ethyl (C_2H_6O)	32		0.547				
	104	0.794	0.648		−180.0		173.0
	176		0.769				
Alcohol, methyl, CH_4O			0.60				
Beeswax, yellow		0.960	0.45	50.0	144.0	144.0	
Beeswax, white		0.965			146.0	146.0	
Beef tallow, fresh		0.968			91.4	109.4	
Beef tallow, old					93.0	110.5	
Glycerine		1.26	0.576				554.0
Mutton tallow, fresh		0.92			97.0	116.6	
Benzol, C_6H_6	22 [Solid]		0.313				
	59	0.885	0.340	54.2			
	104		0.423			42.0	176.0
	149		0.482				
Butter, fresh	212	0.866	0.55		66 to 68	88.0	
Cocoa butter		0.89 to 0.91			69.0	93.0	
Cocoanut oil					68.5	76.0	
Nutmeg butter					91.4	111.0	
Glue		1.27					
Hexane [C_6H_{14}]	32	0.658					
Leaf lard		0.92 to 0.94	0.60	90.0			156.0
Milk, cows		1.028 to 1.035					
Naphtha		0.85	0.45				
Naphthalin, solid, $C_{10}H_8$	50 to 68	1.152	0.314				
	140		0.334	64.8			
Naphthalin, liquid, $C_{10}H_8$	200	0.977	0.41				
Nitric acid		0 661					
Oil, petroleum	100 to 200	0.79 to 0.82	0.50				392 to 482
		0.88	0.51				
Oil, crude, Pa.		0.81	0.50				
Oil, crude, Japan		0.86	0.45				
Oil, crude, Russian		0 91	0 43				
Oil, crude, California		0 96	0.40				
Oil, liquid base		0.83 to 0.87	0 40				
Oil, lubricating		0.83 to 0.87	0 40				590 to 660
Oil, cottonseed			0 47				
Oil, olive	44	0 91	0.47				
Oil, linseed		0 93 to 0.94	0 31		−[illegible].0		600
Oil, turpentine, $C_{10}H_{16}$	−4.0		0.38				
	32.0		0 41				
	176	0.87	0.48				
	320		0.51				320.0
Oil, castor			0.43				
Oil, palm, soft		0.905			69.8	86 0	
Oil, palm, hard					75.2	100.4	
Oil, palm (old)					100.4	107 6	
Paraffin, soft		0.87 to 0.88	0.70	63 0		100 to 125	660 to 730
Paraffin, hard		0.88 to 0 93				130.0	750 0
Spermaceti		0.88 to 0.94		46 4		112.0	
Water, sea		1.026	0.938		27.5		

stances. Unfortunately, these values vary considerably unless the substance is a fixed compound.

The Manufacture of Gasoline.—There is no definite compound called gasoline, but the term is applied to a number of hydrocarbons including propane, butane, hexane, nonane, etc., including a little ethane, all mixed (or dissolved) together. The separation of these hydrocarbon compounds from gas by the means of refrigeration can be accomplished in the case of either casing head or natural gas. The freezing of the gasoline may be carried out by either of two methods. First, there

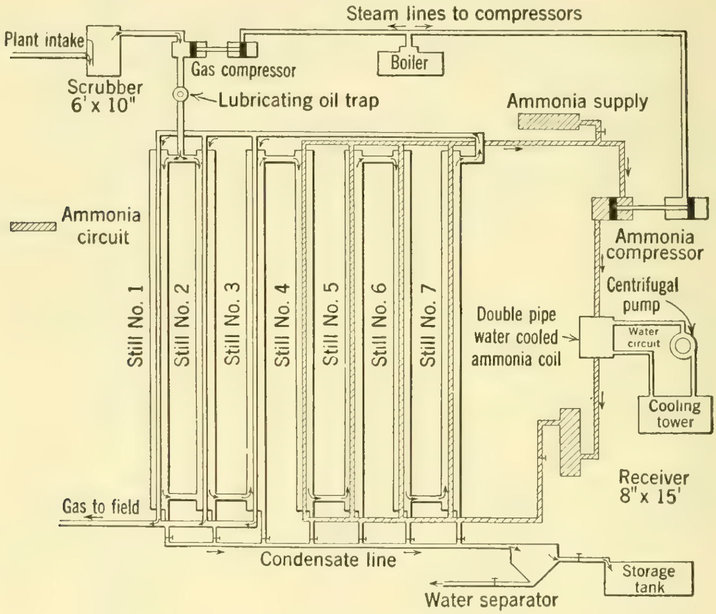


FIG. 374.—Gasoline Manufacture.

is the method by which the gas is compressed (to 160 lb. or over) and then is allowed to throttle down or expand adiabatically in the expansion cylinder of an engine. The temperature is lowered in both of these methods but to a much greater degree in the case of the expanding cylinder, and the gasoline fractions are condensed out during the fall of the temperature. A third method is when the gas is chilled by the use of a refrigerating machine and a suitable exchanger.

Raw casing head gas emanating from producing oil wells is pumped by vacuum pumps to a central point at which it is picked up by a compressor and compressed to a pressure varying from 5 to 40 lb. or more. After this operation the gas passes through a series of water cooled

atmospheric condensers, during which process the heavier gasoline fractions contained in the raw casing head gas are condensed and collected in an accumulator tower 30 to 90 ft. high filled with tile, wooden baffles, lath cuttings, or other special devices manufactured for the purpose of giving a large amount of surface for the absorbent oil to spread over in the downward course through the tower. The oil in the thin film state coming into contact with the upward passing gas absorbs the gasoline content of the gas. This oil passes from the bottom of the tower into the heat exchanger, thence to the pump, which forces it through some water cooled atmospheric coils, thence into a still where the gasoline content is driven off by heat. The denuded oil then passes

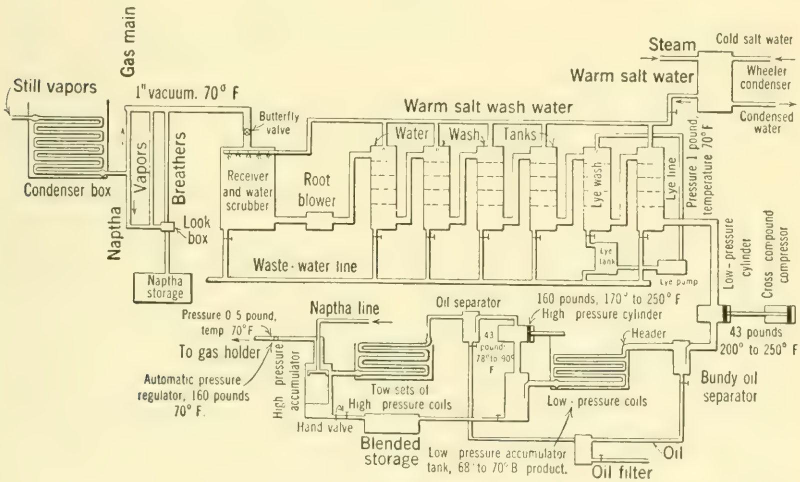


FIG. 375.—Gasoline Manufacture.

into the heat exchanger, then to a pump which forces it through some water cooled atmospheric coils, and finally to the top of the tower from where it began, thus completing the cycle. In the case of natural gas the conditions are the same, practically, except for the characteristics of the gas.

In the case of the absorption process in order to obtain the maximum efficiency of extraction it is necessary to pump a large amount of oil through the tower. This results in a low saturation of the mineral seal oil, a large capacity heat exchanger, a large still and a large pump. Even then it is sometimes impossible to extract the entire gasoline content of the gas in summer time. This is particularly true of plants treating so-called dry or natural gas containing from one-tenth to five-tenths of a gallon per 100 cu. ft. of gas.

In these plants it has been found necessary to use refrigeration to cool the oil down to the temperature at which the vapor tension of the gasoline content of the gas is sufficiently overcome to cause the condensation and absorption by the mineral seal oil absorbent. Therefore, this calls for the introduction of machinery. Also, in the operation of the absorption plants the lighter and more volatile gasoline fractions are not condensed as they are driven off from the still, but pass through with what condensate does form in the absorption accumulator tank, from which point they are withdrawn by a compressor that compresses them up to 80 to 100 lb. After leaving this compressor they pass through a series of water-sprayed condenser coils where they are condensed and separated from the entrained methane and ethane.

It has been found that a gasoline produced in an absorption plant does not need blending, the vapor pressure being low enough so as to allow it to be shipped just as it comes from the still. At some plants gas compressors are used to treat the uncondensed still vapors which come through the cooling coils and make rather a " wild " gasoline of the same characteristics as that produced by other compressor plants. The product is often blended with still run gasoline. Other plants treat the uncondensed still vapors in a secondary absorption tower using mineral seal oil or naphtha as the absorbent. In case mineral seal oil is used, it is sent to the still. If naphtha is used the product is sent to market as it comes from the absorption tower.

The refrigerating machine therefore has a place in gasoline production, although it is not used universally, nor is it of equal advantage in every case. Figures 374 and 375 show the methods used in this kind of work.

Brewery Refrigeration.—The brewery was the first important application of refrigeration in the United States, and for a number of years it was the largest user of mechanical refrigeration, having replaced the use of ice and ice and salt in this industry. Since 1919 the industry has suffered very badly in the United States, and many of the 1225 plants requiring 172,800 tons of refrigeration have been modified into other industries or have been closed.

Refrigeration in lager beer brewing is required for the following purposes:

(*a*) To cool the liquid resulting from the malt and hops (wort) from about 70 to 40 deg. F.

(*b*) To cool the wort, at one period or another, from 40 to 32 degrees.

(*c*) To neutralize the effect of fermentation.

(*d*) To cool the cellars, yeast rooms, hop stores, etc., kept at 32 to 38 deg. F.

To cool the wort to 40 degrees requires:

$$Q_1 = B \times W \times d \times C \times (t - 40) \text{ B.t.u.},$$

where

B = the number of barrels;
W = the nominal weight of a barrel (about 360 lb.);
d = the density of the wort (1.05);
c = the specific heat of the wort (0.9).

To cool the wort to 32 degrees requires:

$$Q_2 = B \times 265(40 - 32).$$

To neutralize the heat of fermentation:

The action of fermenting is

$$C_{12}H_{22}O_{11} = 4C_2H_6O + 4CO_2 + 330 \text{ B.t.u. per lb. of maltose.}$$

$$Q_3 = \frac{B \times 0.91 \times (b - b_1)(259 + b) \times 280}{100},$$

where b and b_1 = the specific gravity of the wort and ready beer by Balding saccharometer (about 1.055 and 1.015 respectively).

To cool the cellars, etc.: This is a calculation similar to that in cold storage and depends on the heat leakage, etc., through the building material. The temperatures required are nominal and the result is that the suction pressure on the compressor is usually 30 lb. gage or higher.

Soap Manufacture.—Refrigeration is used quite extensively in the United States in the manufacture of soap. As the applications vary in the different plants examples will be given as representative of common practice.

In the manufacture of "boraxine," a flaky soap, the hot soap arrives at the refrigerating apparatus at about 212 deg. F. The usual chilling device is the roll, revolving at approximately 10 r.p.m. similar to the lard and compound rolls (Fig. 300). When brine is used for cooling it enters at 10 deg. and leaves at about 18 deg. F., and if direct expansion is used the evaporating temperature is about 15 deg. F. By the action of the roll the flaky boraxine is chilled to about 32 degrees and the solid soap is scraped off the surface of the roll by means of a knife. The refrigerating load is similar, except for the physical constants, to that given in Chapter XIV.

In making certain toilet soaps it was the practice at one time to fill large drums containing several hundred pounds of the liquid soap and this was permitted to air cool until the entire mass was solid, but this method was found to be both slow and awkward. The modern bar

method is shown in Fig. 376. This is an endless chain of molds, and means are provided by which the molds are made to pass through the long refrigerating box, the speed of the chain and the length of the box being such as will permit a complete hardening of the soap before a complete circuit can be made. The refrigeration is that due to the heat leakage and the cooling and solidifying of the bars. About 15 lb. gage is usually carried in the evaporating coils in the refrigerated box.

The Manufacture of Rubber.—In the manufacture of rubber, refrigeration is used to assist in the removal of the heat generated in the mill rolls caused by the mixture of rubber with its compounds on the surface of the rolls. The surface speed ratio of the two rolls through

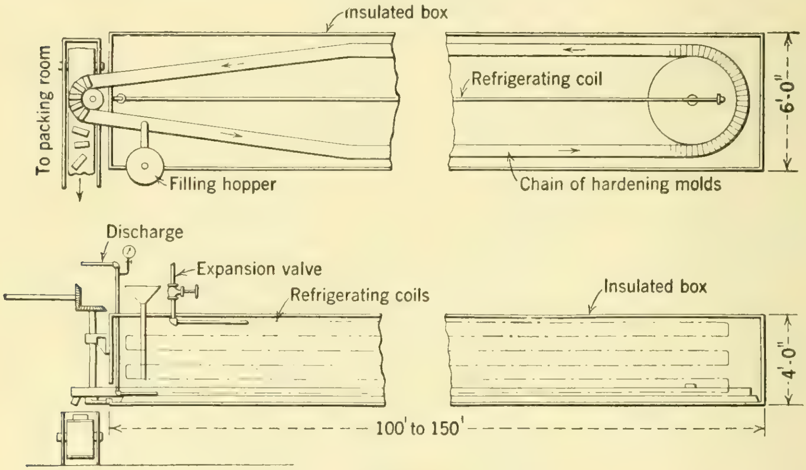


FIG. 376.—Chilling of Bar Soap.

which the rubber passes is 1 : $1\frac{1}{2}$; the variation in speed being required to effect the kneading or milling which is desired. The line shaft is attached to the faster roll, and the rubber sticks to the slower one. The operator cuts the rubber off the roll and returns it to the hopper for more kneading. For this purpose the rolls usually are

16 in. diameter by 42-in. face with shell thickness of 4 in.
20 in. diameter by 60-in. face with shell thickness of 5 in.
24 in. diameter by 84-in. face with shell thickness of 6 in.

Nearly all the power delivered to the rolls tends to heat the rubber. To prevent burning, the roll surfaces are limited to not more than 160 deg. F., and this necessitates a water temperature of 58 degrees or lower entering and 64 degrees or lower leaving the rolls. Roughly, the heat which must

be absorbed by the water is 0.9 of the heat equivalent of the power delivered to the shaft or, roughly, 0.9 hp. per 1 in. of roll face.

Problem.—Tests show that a motor driving a roll requires 30 kw. What refrigeration is required?

$$\text{Refrigeration} = \frac{30 \times 0.85 \times 0.9 \times 1.34 \times 42.4}{200} = 6.52 \text{ tons.}$$

Formerly refrigeration was used for making it possible to cut block rubber into thin sheets, after the block had been frozen. This method is obsolete at the present time.

Gelatine Emulsion.—Large chill rooms are maintained at from 35 to 40 deg. F. for the purpose of chilling emulsion. This converts it from a warm gelatine solution to a solid jelly, and it is best kept in a solid condition for various lengths of time. For photographic work the emulsion is melted at from 90 to 100 deg. F. and coated on a paper or a film base after which it is submitted to a sudden chill. Refrigeration in industry of this nature consists in humidity control and a very careful temperature regulation.

Glue.—Both glue and gelatine solutions are cooled by means of refrigeration to 50 or 40 deg. F. The glue liquid, containing approximately 25 per cent dry glue, is sometimes chilled in small pans containing 72 lb. (6 in. by 9 in. by 36 in.), or the glue solution may be fed on to belts (Fig. 324), so arranged that the proper temperature will be attained before the solution leaves the chill chamber.

Refrigeration in Gas Warfare.—Refrigeration in gas warfare[2] or in general in the preparation of poison and some noxious gases may involve:

(*a*) The cooling of the gas.

(*b*) The liquefying of the gas.

(*c*) The cooling of the liquid.

(*d*) The cooling of the metal containers required for the storage of the condensed gas to a temperature below 32 deg. F.

In the case of phosgene gas, formed by bringing equal volumes of CO and Cl_2 together in the presence of a catalyzing agent, the gas was cooled in a double pipe water cooler, then cooled and condensed by passing through helical lead coils submerged in brine where liquefaction occurred at 40 deg. F. Mustard oil is stated to require 2 tons of refrigeration per one ton of oil per day of 24 hours.

The Haber Process.—Ammonia is now being made by the fixation of atmospheric nitrogen by the Haber process at 100 to 200 atm. pressure, using a mixture of water and producer gas; the Casale, at 600 to

[2] A. M. Heritage, Journal Amer. Soc. Refrigerating Engineers, May, 1919.

700 atm., using a mixture of electrolytic hydrogen and nitrogen obtained by burning the oxygen from the air with excess hydrogen; and the Claude, at 900 atm. pressure exerted on a mixture nitrogen from the liquid air process and hydrogen from coke oven gas. One plant, at Syracuse, N. Y., is working on a modified Haber process similar to that used by the United States Government Nitrate Plant No. 1 during the World War, and has a capacity of 10 tons per day (1925). According to Frederick Pope,[3] the cost of synthetic ammonia can be reduced to 4 cents per lb. in large plants. The following method has been used in the calculation of the required refrigerating effect in the Haber process.

Problem.—90,000 cu. ft. per hour of permanent gases ($3H_2 - N_2$) at 60 deg. F. and 1 atm., under 100 atm. and 10 per cent ammonia content leaving the catalyst, is to be cooled in two coolers, one to 10 deg. and the other to − 40 deg. F. The liquefied ammonia will be trapped off after each cooler. The value for c_p for the permanent gases at 100 atm. is to be taken as 1.0. Find the refrigeration required. For gas mixtures

$$p = p_1 + p_2 = 1470 \text{ lb.}$$

$$= p_{\text{saturated ammonia}} + p_{\text{permanent gas}},$$

but

$$\frac{V_1}{V} = \frac{p_1}{p} = 0.1;$$

therefore,

$$1470 = 147 + 1323,$$

and

$$t_{NH_3 \text{ at } 147 \text{ lb.}} = 77.63 \text{ deg. F.}$$

Liquefaction.

At 10 degrees $\dfrac{V_1}{V} = \dfrac{p_1}{p} = \dfrac{38.51}{1470} = 2.62$ per cent by volume.

At −40 degrees $\dfrac{V_1}{V} = \dfrac{p_1}{p} = \dfrac{10.41}{1470} = 0.709$ per cent by volume.

Volume of Permanent Gases.

At 100 atm. and 77.6 deg. F.:

$$90{,}000 \times \frac{14.7}{1323} \times \frac{460 + 77.6}{460 + 60} = 1035 \text{ cu. ft.}$$

At 100 atm. and 10 deg. F.:

$$90{,}000 \times \frac{14.7}{1431.5} \times \frac{460 + 10}{460 + 60} = 833 \text{ cu. ft.}$$

[3] Frederick Pope, Ammonia Synthesis, Refrigerating Engineering, 1926.

At 100 atm. and −40 deg. F.:

$$90{,}000 \times \frac{14.7}{1460} \times \frac{460 - 40}{460 + 60} = 732 \text{ cu. ft.}$$

Weight of Ammonia Present (lb.)

At 77.6 deg. F. Total volume ÷ specific volume of NH_3 $= \frac{1033}{2.034} = 508$ lb.

At 10 deg. F. $= \frac{833}{7.304} = 114$ lb.

At −40 deg. F. $= \frac{732}{24.86} = 29.4$ lb.

Quality of Vapor Leaving Refrigerating Coils.

At 10 deg. F. $114.0 \div 508 = 22.45$ per cent (by weight).

At −40 deg. F. $29.4 \div 114.0 = 26.3$ per cent (by weight).

Weight of Permanent Gases.

Taking the specific volume, $3H_2 - N_2 = 44.24$ at NTP,

$$\frac{90{,}000}{44.24} = 2034 \text{ lb.}$$

Ammonia Content (by weight).

At 77.6 deg. F. $\frac{508}{2034 + 508} = 20.1$ per cent.

At 10 deg. F. $\frac{114}{2034 + 114} = 5.33$ per cent.

At −40 deg. F. $\frac{29.4}{2034 + 29.4} = 1.43$ per cent.

Refrigeration.—As the cooling takes place at constant pressure, the heat absorbed by the evaporating coils is equal to the difference in the value of i.

$$Q = i_{1_{gas}} + i''_{1_{Ammonia}} - (i_{2_{gas}} + i'_{2_{NH_3}} + x_2 r_{2_{NH_3}})$$

$$= 2034 \times 1.0 \times [77.6 - 10] + 508\ [(630.3 - 53.8) - (0.225 \times 561.1)]$$

$$= 366{,}300 \text{ B.t.u.} = 30.5 \text{ tons.}$$

Likewise, the refrigeration in the second stage cooler at a temperature of −40 deg. F. becomes:

$$Q = (2034 \times 1.0 \times 50) + (614.9 - 0 - 0.263 \times 597.6)$$

$$= 167{,}600 \text{ B.t.u.} = 13.95 \text{ tons.}$$

In the first stage the evaporating pressure, using ammonia as the refrigerant, would be at 15 lb. gage, whereas it would be at 8 lb. absolute in the second stage. The percentage of ammonia in the gas phase as found experimentally is not exactly as given in the calculations, but this method is given in full as an example of what can be done.

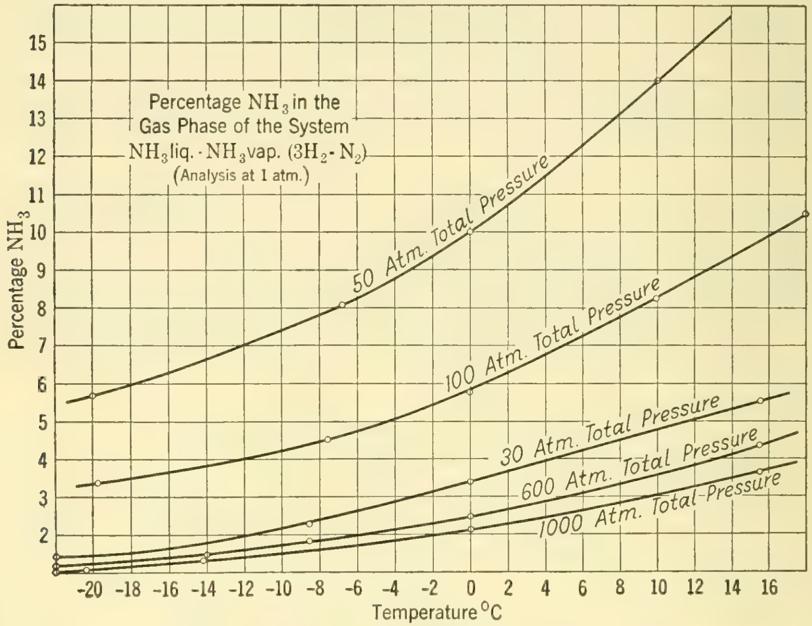


Fig. 377.—Synthetic Manufacture of Ammonia.

Figure 377 gives more accurately the percentage of ammonia in the gas phase of the system $3H_2 - N_2$. This was developed by the fixed nitrogen section of the Department of Commerce. The calculation is given more as a method than as a means of arriving at an accurate answer.

CHAPTER XIX

SAFETY DEVICES AND FIRE PROTECTION

Because of the nature of the refrigerants used, the operating cycle, and the pressures encountered, a number of special regulations have been devised for refrigerating plants. These regulations consist in means for prevention of excess pressures in the discharge lines from the compressor, the condenser or the liquid receiver, or (in special cases, as in the case of fire) of discharging the whole of the refrigerant into the atmosphere or a suitable mixing chamber.

The Safety Valve.—The safety valve for ammonia is simply a pop valve ground to its seat, whereas the carbonic valve has a thin metal disc usually in addition to the pop valve. The valve is best piped to discharge into the low-pressure side of the system and should be placed on the shell type brine cooler, the shell type condenser, and the liquid receiver where the pressure can become excessively high on the occasion of the temperature rising with both the inlet and the outlet valves closed. If a pop valve blows, it must be reground in order to be put back into service. When the discharge is into a mixer (Fig. 381), or into the atmosphere, the refrigerant is lost.

When the compressor is *electrically driven* it is usual to have special devices to stop the compressor should the pressure into the discharge line rise to an excess: for example, as high as 250 lb. in the case of ammonia. In Fig. 378 the gas pressure exerted on the diaphragm compresses a spring and moves a plunger, thereby releasing a spring operated trip which is connected to a knife switch. Various valves of similar design are manufactured.

Should a cylinder head break, or similar trouble develop, a non-return valve comparable to those used in steam boiler headers would be very valuable, and such valves are installed in large plants in order to prevent the loss of the entire charge and the consequent vitiation of the compressor room and the vicinity. A hand-operated valve is shown in Fig. 379, and this can be modified to be operated by means of a push button. In either case a remote operation of the stop valve is possible.

Helmets.—Every plant above the fractional tonnage size should be provided with suitable gas masks or helmets in good working order.

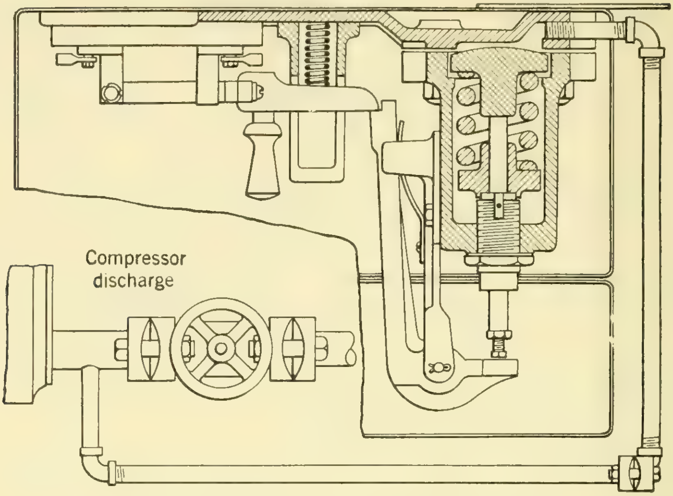


FIG. 378.—Pressure Actuated Cut Out.

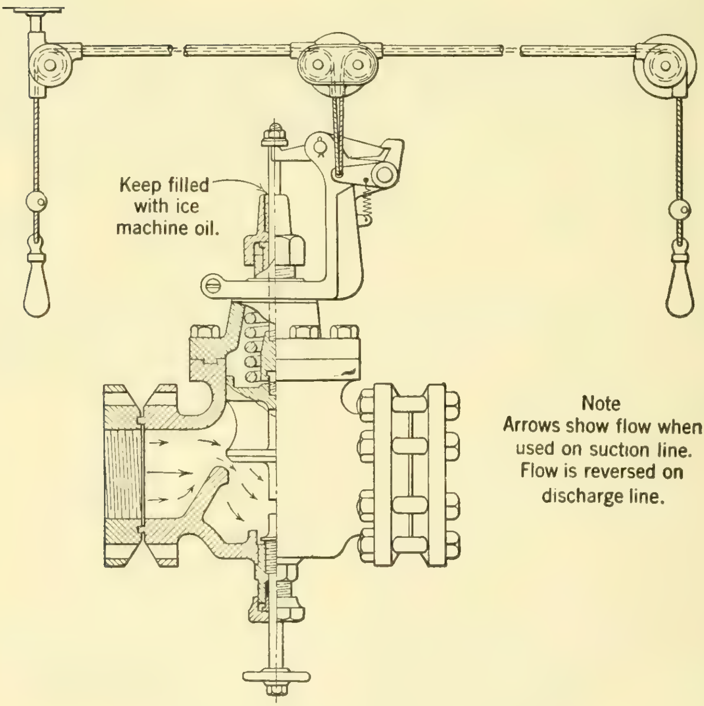


FIG. 379.—Remote Control Stop Valve.

They should be placed *outside* the compressor room, but within easy access at all times.

Open Flames.—With ammonia, ethyl and methyl chloride and the hydro-carbon refrigerants no flames, arc lights, gas jets or any apparatus employing a flame except internal combustion engines with internal ignition should be permitted. It will be seen in Chapter VII that the refrigerants listed are explosive with certain limited mixtures and, as a matter of fact, several bad explosions have been laid to the door of ammonia.

Inspection of Refrigerating Plants.—The City of Chicago inspection rules for refrigerating plants requires the inspector to note particularly the following 10 points:

1. Valves and fittings entering the construction of machinery, apparatus, pipes and the connections therewith, and wherein ammonia is used under pressure, shall be subjected to an air pressure test of 300 lb., and 500 lb., of hydrostatic pressure.
2. Hangers and braces shall be of flat angle or round wrought iron or steel.
3. No perforated material will be accepted.
4. Supports and brackets may be of cast iron.
5. The discharge pipe between compressor or generator and condenser sections shall be extra heavy material and be provided with two stop valves.
6. All liquid pipes must be of extra heavy material.
7. High and low-pressure check valves, or equivalent, must be located as close to compressor as convenient, and provided with a by-pass. Check valves in absorption plant shall be located between rectifier and condenser and in the discharge line close to the aqua ammonia pump.
8. Safety pop valve must be located between compressor or generator and first stop valve and arranged for seal and set at 250 lb. for ammonia compressors and 1500 lb. for CO_2 compressors.
9. Size of safety valve for 250 lb. pressure:

    Capacity not exceeding 10 tons, $\frac{1}{2}$ in.
    Capacity over 10 tons and not exceeding 25 tons, $\frac{3}{4}$ in.
    Capacity over 25 tons and not exceeding 40 tons, 1 in.
    Capacity over 40 tons and not exceeding 60 tons, $1\frac{1}{4}$ in.
    Capacity over 60 tons and not exceeding 100 tons, $1\frac{1}{2}$ in.
    Capacity over 100 tons and not exceeding 140 tons, 2 in.
    Capacity over 140 tons and not exceeding 190 tons, $2\frac{1}{2}$ in.
    Capacity over 190 tons and not exceeding 300 tons, 3 in.
    Tonnage based on 7,500 cu. in. displacement per ton for ammonia compressor.

10. Separators, liquid receivers and receptacles on the high pressure side of an ammonia machine, shall be tested to a hydrostatic pressure of 500 lb. and 300 lb. of air. Hydrostatic test to be stamped on each vessel with name of maker, shell brine coolers, purifiers, absorbers, carrying a maximum working pressure below 250 lb., shall be equipped with a safety valve not less than $\frac{1}{2}$ in. diameter, and set for a pressure arrived at by computing the material and construction of each apparatus according to the rules applied thereon. Liquid receivers must be equipped with a $\frac{1}{2}$ in.

pop safety valve, and a stop valve between safety valve and receiver. Arrange stop valve so it can be sealed when open. Outlet of safety valve to be piped to atomsphere outside of building.

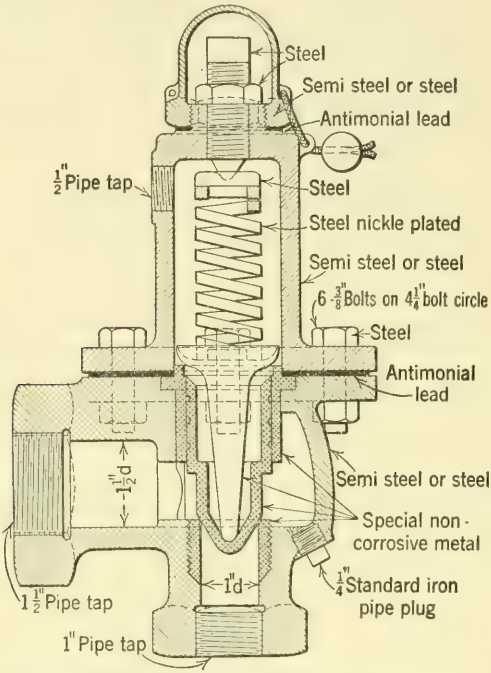


Fig. 380.—Pop Safety Relief Valve.

The Mechanical Refrigerating Safety Code.—A safety code for refrigerating machinery has been devised by the American Society of Mechanical Engineers in conjunction with the American Society of Refrigerating Engineers (1925),[1] under the direction of the American Engineers Standards Committee, which is as follows:

MECHANICAL REFRIGERATION SAFETY CODE [2]

SECTION 10

Scope

This code shall apply to every refrigerating equipment employing fluids which are vaporized and liquefied or compressed in their refrigerating cycle and is intended

[1] See also the A.S.M.E. Boiler Code, Part I, Section 6, for rules for the construction of unfired pressure vessels.

[2] Formulated under direction of American Engineering Standards Committee and sponsorship of The American Society of Refrigerating Engineers.

to provide for the safe installation, operation and inspection of refrigerating systems, as well as the storage of refrigerants.

Interpretations and Exceptions

The purpose of this code is to provide reasonable safety for life, limb, health and property. It shall be liberally construed to secure these results by the enforcing officers or body, who shall have authority in cases of practical difficulty or unnecessary hardship to grant exceptions from the literal requirements of this code, as long as equivalent protection is thereby secured. When the safeguarding of specific types of machines is covered by other approved codes, these codes shall be given preference. When specific devices or methods are mentioned in this code, other devices or methods which will secure equally good results may be used, subject to the approval of the administrative authority.

SECTION 20

Definitions

(*a*) *Refrigerating Equipment.*—A "refrigerating equipment" or "refrigerating system" is an apparatus, including its source of energy in so far as it affects its safety of operation, for abstracting heat from a body at one temperature and transferring it to another at a higher temperature.

(*b*) *Capacity.*—The "capacity" of a refrigerating equipment is its rating in tons of refrigeration per twenty-four (24) hours, determined as follows:

a′. A standard ton of refrigeration is two hundred and eighty-eight thousand (288,000) B.t.u.

b′. The standard commercial ton of refrigeration is at the rate of two hundred (200) B.t.u. per minute.

c′. The standard rating of a refrigerating machine, using liquefiable vapor, is the number of standard commercial tons of refrigeration it performs under the following adopted refrigerant pressures:

(1) The inlet pressure is that which corresponds to a saturation temperature of five degrees (5°) F. (−15° C.).

(2) The outlet pressure is that which corresponds to a saturation temperature of eighty-six degrees (86°) F. (30° C.).

These pressures are measured outside and within ten (10) feet of the pressure imposing element, measured along the inlet and outlet pipes, respectively.

(*c*) *Working Pressure* shall be understood to mean the maximum allowable working pressure.

(*d*) *Pressure Imposing Element* is that part of a refrigerating equipment which draws the refrigerant from the low-pressure side and discharges it into the high-pressure side of the system, such as a compressor or absorber and generator.

(*e*) *Machinery Room* is one in which is located any pressure-imposing element, condenser, receiver or shell-type apparatus. Rooms which contain only pipes for conveying the refrigerant are not considered machinery rooms.

(*f*) *Pressure Limiting Device* is a device acting directly responsive to pressure to stop the action of the pressure-imposing element.

(*g*) *Pressure Relief Devices*—

a′. *Pressure Relief Valve* is a device that will automatically relieve the excess pressure at a predetermined maximum difference of pressure.

b'. *Automatic By-Pass* is a device that will automatically relieve within narrow limits of a predetermined pressure the excessive pressure, irrespective of the outlet pressure.

c'. *Fusible Plug* is a device which will, by the fusing or softening of a suitable material, relieve the pressure when the device has reached a predetermined temperature.

d'. *Rupture Member* is a device that will rupture at a predetermined difference of pressure.

(*h*) *Vessel* is any part of a system containing refrigerant.

(*i*) *Liquid Receiver* is a liquid refrigerant reservoir permanently connected by inlet and outlet pipes to a refrigerating system.

(*j*) *Container* is a cylinder or drum for the shipment of a refrigerant.

(*k*) *Suction and Discharge Stop.*—The terms "main suction stop" and "main disharge stop" designate the manually operated shut-off valves located in the suction and discharge lines, respectively, adjacent to the pressure imposing element.

(*l*) *Service Valve* is a valve located in the refrigerant circuit which may be operated only by a specially constructed service key.

SECTION 30

General Requirements

(*a*) Pressure relief and pressure limiting devices shall be made of material suitable for the refrigerant employed and, where practicable, their working parts shall be non-corrodible and shall be set, marked and sealed by the manufacturer.

(*b*) Automatic by-passes shall be of such design as to operate within five per cent (5%) of the pressure for which stamped, irrespective of the outlet pressure.

(*c*) If rupture member devices are used, they shall be rupturable by a direct pressure. The rupturing pressure shall be legibly marked on the rupturing member or on proper attachment thereof.

(*d*) There shall be no valve between any pressure relief devices and the receivers, condensers, evaporators, or other vessels which they protect or between automatic by-pass devices and the main suction line. There shall be also no valve between any rupture member or fusible plug or other pressure limiting means and the part of the equipment protected thereby, except that service valves may be permitted on Class C Equipment.

(*e*) All piping, and liquid receivers containing the refrigerant, shall be so installed as to be least liable to damage in case of accident.

(*f*) Electrical equipment shall be installed in accordance with the National Electrical Code.

(*g*) In Class A and Class B equipment there shall be a service switch, or remote control switch, controlling all the electrically driven refrigerating machinery, one control of which, properly labeled, shall be located outside of the machinery room, where it can be quickly and safely reached and operated in case of an emergency.

(*h*) Switchboards and high tension cables shall be located where they will be least liable to damage if an accident occurs to the refrigerating equipment.

(*i*) All working parts of refrigerating machines, prime movers and drive belts shall be guarded in accordance with the provisions of the National Safety Code for Mechanical Transmission of Power.

SECTION 30

Operating Precautions

(*a*) Containers having refrigerant therein shall be stored in a cool, well ventilated place, as remote as possible from danger by heat or fire.

(*b*) Refrigerant not contained in the refrigerating system shall be stored only in containers conforming with the regulations of the Interstate Commerce Commission for the transportation of such refrigerant.

(*c*) When refrigerant is withdrawn from the system, it shall be discharged to the outside atmosphere or suitable absorbent or stored in containers conforming with the regulations of the Interstate Commerce Commission for the transportation of such refrigerant. Care shall be taken to prevent escape of refrigerant into building.

(*d*) In withdrawing refrigerant from the equipment into containers, they shall be placed on a scale and shall not be filled to more than allowed capacity as prescribed by Interstate Commerce Commission regulations.

(*e*) Containers shall not be connected to the system except during period of charging.

(*f*) Repairs to refrigerant lines or apparatus shall not be made while they are under pressure. Tightening of bolts or flanged joints shall be done at a reduced pressure.

(*g*) In testing systems with air pressure care shall be taken to prevent the temperature at any point rising above one hundred and thirty degrees (130°) F. (hand warm).

Part I—Ammonia Systems

SECTION 100

General Requirements for Ammonia Systems

(See also General Requirements, Section 30.)

(*a*) Every part of the high pressure side of the system shall be designed for a maximum allowable working pressure of at least two hundred (200) pounds per square inch gauge pressure and every part of the low-pressure side shall be designed for a maximum allowable working pressure of at least one hundred and fifty (150 pounds) per square inch gauge pressure. A factor of safety of five (5) shall be used in all designs, except for gauges and control mechanism.

(*b*) Every vessel in the system, except connecting piping, shall be marked with the maximum allowable working pressure and shall be tested hydrostatically to one and one-half ($1\frac{1}{2}$) times this pressure marking, such tests to be conducted in accordance with the "Unfired Pressure Vessel Code."

SECTION 110

Class A Equipment

A "Class A Equipment" is a refrigerating system of thirty (30) tons capacity or over, or containing one thousand (1000) pounds of ammonia or over.

Rule 1100.—*Pressure Imposing Element.*

(*a*) Every equipment shall be provided with one or more automatic by-passes of proper size, connected between each main discharge stop valve and the pressure

imposing element, to relieve excessive pressures into the low pressure side, on either side of the main suction stop valve; or, as an alternative, shall be provided with one or more pressure relief valves of proper size discharging, during the period of excess pressure, to the outside atmosphere (see Rule 1103 (*a*)), or to the low-pressure side when a pressure relief valve is installed thereon discharging to the outside atmosphere. These pressure relief devices shall be so designed and installed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure. The proper (nominal pipe) sizes of such devices shall be as follows:

For equipment of from	30 to 60 tons capacity, use at least one $\frac{3}{4}$ in. device.
	60 to 100 tons capacity, use at least one 1 in. device.
	100 to 175 tons capacity, use at least one $1\frac{1}{4}$ in. device.
	175 to 250 tons capacity, use at least one $1\frac{1}{2}$ in. device.
	250 to 450 tons capacity, use at least one 2 in. device.
	450 to 900 tons capacity, use at least two 2 in. devices.

(*b*) Every equipment shall have an automatic pressure-limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the pressure relief valve.

(*c*) A cushioned or other satisfactory check valve shall be placed in the discharge line between the pressure-imposing element and the condenser.

Rule 1101.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch automatic by-pass set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point to discharge to that part of the low-pressure side of the equipment protected by an emergency relief device. As an alternative, a pressure relief valve discharging to the atmosphere may be used. (See Rule 1103 (*a*).)

Rule 1102.—*Fire Emergency Devices.*

(*a*) The low-pressure side of every equipment shall be provided with a hand-operated relief valve for discharging the ammonia in case of fire, either to the atmosphere, to a suitable body of water or through a mixer.

(*b*) The hand-operated relief valve shall be located on the outside of the wall of the engine room, or shall be capable of operation therefrom. The valve or operating mechanism shall have a protecting cover.

(*c*) The minimum (standard pipe) sizes of the hand relief valves shall be as follows:

Where the charge is	1,000 to 1,800 lb., use one $\frac{3}{4}$ in. valve.
	1,800 to 3,000 lb., use one 1 in. valve.
	3,000 to 5,250 lb., use one $1\frac{1}{4}$ in. valve.
	5,250 to 7,500 lb., use one $1\frac{1}{2}$ in. valve.
	7,500 to 13,500 lb., use one 2 in. valve.
	13,500 to 27,000 lb., use one 3 in. valve.

Rule 1103.—*Refrigerant Discharge.*

(*a*) When discharged into the atmosphere, the ammonia shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the ammonia with air. The piping from the hand relief valve to

point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the valve. Such piping shall be used only for ammonia discharge and be so located that the gases discharged are not liable to become ignited. Outlets from diffusers shall be above the roof of any building within fifty (50) feet.

(*b*) Outlets from atmospheric pressure relieve devices shall connect to the discharge pipe specified in Paragraph (*a*) of this rule, or a similar pipe.

(*c*) In localities where the discharge of a refrigerant in case of fire comes under the direction of the fire department, the hand relief valve shall be located in a locked box, which can be opened only by members of the fire department by means of a fire department key. The door on this box shall be on the public thoroughfare side of the building wall. If this is impracticable, the door shall be in a vestibule having glass panel doors, which provide easy access from the street. The door of the box shall be at a height of not more than five (5) feet nor less than two (2) feet above the street or vestibule floor level. In either case it shall be in a place as remote as possible from any exhaust vent or outlet from the machinery room. At the discretion of the fire department, such box may be equipped with an auxiliary door, also locked, to be opened only by such persons as may be duly authorized by the fire department to make repairs.

(*d*) The protecting box shall be plainly labeled "For Fire Department Use Only." The emergency valve shall be labeled "Ammonia." In case the ammonia is to be discharged into the sewer, as provided in Paragraph (*e*), there shall be also within the box a sign reading "Do not open valve until water is flowing into the mixer."

(*e*) When the local fire department requires that a refrigerant be discharged through a mixer into the sewer, there shall be provided a check valve and a standard fire department connection through which the fire department supplies the necessary water under the proper pressure. A stop valve shall be installed inside the building in the refrigerant line leading to the mixer hand relief valve for use in repairing same. This valve shall be sealed open, except for and during repairs.

(*f*) The fire department shall make at least an annual inspection of valves, emergency means and mixer.

Rule 1104.—*Helmets or Masks.*

(*a*) At least two (2) helmets or masks of approved make, tested in accordance with the requirements of the United States Bureau of Mines for ammonia, shall be kept in operative condition and available for the immediate use of responsible attendants, who shall be trained in their use.

(*b*) Helmets or masks shall be kept in a place easily accessible from outisde the machinery room.

Rule 1105.—*Open Flames.*

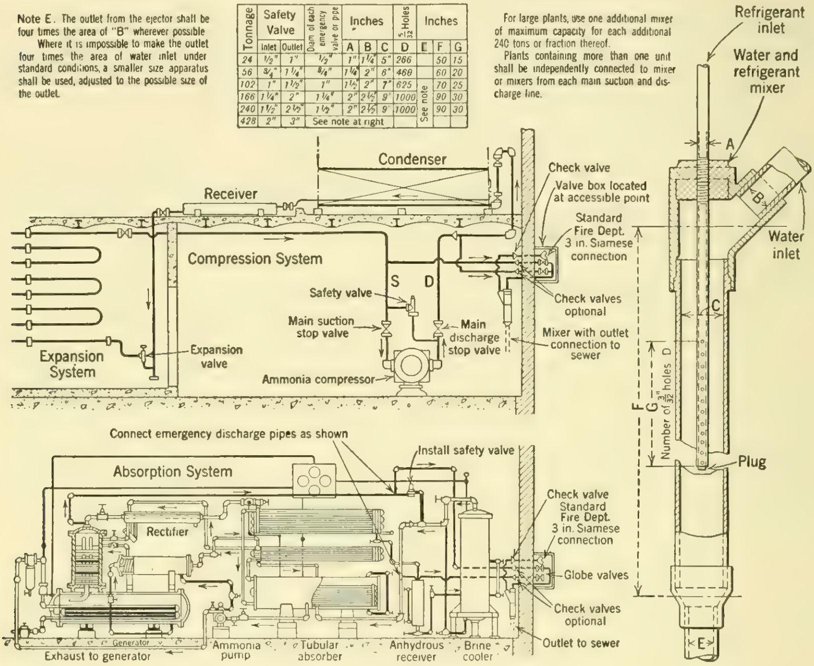
(*a*) In machinery rooms, there shall be no fire, arc light, hot surface ignitor or flame open to the atmosphere of the room; any device intended to adequarely isolate the fire, arc light, hot surface ignitor or flame shall have been approved by a competent laboratory and the administrative authority. Internal combustion engines with hot surface ignition may be started in the usual manner.

Rule 1106.—*Electrical Equipment.*

(*a*) All current carrying devices, including bus bars, operating at above six hundred (600) volts shall have all live parts in tight enclosures, or be provided with suitable insulating covering.

(*b*) Extensive electrical equipment other than that essential to the ice making or refrigerating plant shall not be permitted in the machinery room.

(*c*) Transformers with primary or secondary connections above six hundred (600) volts shall be in locked vaults which, if adjoining the machinery room, shall be separated therefrom, preferably by unpierced walls, ceilings and floors of six (6) inches of reinforced concrete or its equivalent. All conduit ends and wall openings for conduits entering the vault shall be sealed. (See also the National Electrical Code.)



Note E. The outlet from the ejector shall be four times the area of "B" wherever possible. Where it is impossible to make the outlet four times the area of water inlet under standard conditions, a smaller size apparatus shall be used, adjusted to the possible size of the outlet.

Tonnage	Safety Valve		Diam of each emergency valve or pipe	Inches			$\frac{5}{32}$ Holes	Inches		
	Inlet	Outlet		A	B	C	D	E	F	G
24	1/2"	1"	1/2"	1"	1 1/4"	5"	266		50	15
56	3/4"	1 1/4"	3/4"	1 1/4"	2"	6"	460		60	20
102	1"	1 1/2"	1"	1 1/2"	2"	7"	625	See note	70	25
166	1 1/4"	2"	1 1/4"	2"	2 1/2"	9'	1000		90	30
240	1 1/2"	2 1/2"	1 1/2"	2"	2 1/2"	9'	1000		90	30
428	2"	3"	See note at right							

For large plants, use one additional mixer of maximum capacity for each additional 240 tons or fraction thereof.

Plants containing more than one unit shall be independently connected to mixer or mixers from each main suction and discharge line.

FIG. 381.—Ammonia Mixer Valve.

Rule 1107.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser, or high-pressure side, and one for indicating pressure in evaporator, or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

Rule 1108.—*Exits, Ventilation.*

(*a*) Each machinery room shall have a direct exit to the open air, or to a room or hall from which gases from machinery room can be excluded by close fitting self-closing door.

(*b*) Vertical and horizontal openings that permit the passage of gases to other parts of the building, which are not a part of the refrigerating plant, shall be sealed or provided with close fitting self-closing door.

(*c*) Each machinery room shall be provided with means for adequate ventilation to the open air, either direct or by means of a suitable duct or ducts. Where a mechanical system of ventilation is employed, the control of such system shall be easily accessible and located outside of the machinery room.

Rule 1109.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low pressure side.

SECTION 120

Class B Equipment

A "Class B Equipment" is a refrigerating system of less than thirty (30) tons capacity, or containing less than one thousand (1000) pounds and over fifteen (15) pounds of ammonia.

Rule 1200.—*Pressure Imposing Element.*

(*a*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall be provided with a one-half ($\frac{1}{2}$) inch automatic by-pass connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressure into the low-pressure side on either side of the main suction stop valve; or as an alternative through a one-half ($\frac{1}{2}$) inch (standard pipe) pressure relief valve discharging as specified in Rule 1203 (*a*), or to the low-pressure side. These pressure relief devices shall be so designed and installed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure.

(*b*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall have an automatic pressure-limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the relief valve.

Rule 1201.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch automatic by-pass, set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point, to discharge to that part of the low-pressure side of the equipment protected by an emergency relief device. As an alternative, a pressure relief valve discharging to the atmosphere may be used. (See Rule 1203 (*a*).)

Rule 1202.—*Fire Emergency Devices.*

(*a*) Every equipment shall be provided with a one-half ($\frac{1}{2}$) inch pressure relief valve connected to the low-pressure side of the equipment, set to relieve excessive pressure in a manner specified in Rule 1203 (*a*), and shall discharge at a pressure not higher than the maximum allowable working pressure.

(b) Class B Equipment containing less than one hundred (100) lb. of ammonia and so designed or provided with devices as to prevent the existence of pressures in any part of the equipment in excess of the maximum allowable working pressure may be protected by one pressure relief valve.

(c) Class B Equipment containing less than one hundred (100) lb. of ammonia, in which the design of the pressure-imposing element is such that the generation of pressure in excess of the maximum allowable working pressure is impossible, may be protected by fusible plugs on each part of the equipment containing ammonia which can be exposed to heat from an external source, provided pressure generated at the temperature necessary to fuse the plug is less than the maximum working pressure.

Rule 1203.—*Refrigerant Discharge.*

(a) The ammonia passed by the pressure relief devices shall be discharged either to the atmosphere or to a suitable body of water.

(b) When discharged into the atmosphere, the ammonia shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the ammonia with air. The piping from the pressure relief device to point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the device. Such piping shall be used only for ammonia discharge and be so located that the gases discharged are not liable to become ignited. Outlets from diffusers shall be above the roof of any building within fifty (50) feet.

(c) When the ammonia is discharged into a tank of water, the tank shall not be used for any other purpose. At least one (1) gallon of fresh water for every pound of ammonia contained in the equipment shall automatically be maintained in the tank. Provision shall be made to prevent the water from freezing without the use of salt or other chemicals.

(d) The tank shall be substantially constructed with a hinged cover or, if it is of the enclosed type, it shall have both a water inlet and a vent hole at the top. No horizontal dimensions of the tank shall be greater than one-half ($\frac{1}{2}$) the height. The discharge pipe from the pressure relief devices shall be so attached as to discharge the refrigerant at the center of the bottom and that portion of the pipe within the tank shall be of lead with a one-sixteenth ($\frac{1}{16}$) inch vent hole above the water level. The tank shall be securely supported and firmly braced. There shall be no opening in the tank below the water level and the water level shall be at least six (6) inches below the top of tank.

Rule 1204.—*Helments or Masks.*

(a) At least one (1) helmet or mask of approved make, tested in accordance with the requirements of the United States Bureau of Mines for ammonia, shall be kept in operative condition and available for the immediate use of responsible attendants, who shall be trained in their use.

(b) Helmets or masks shall be kept in a place easily accessible from outside the machinery room.

Rule 1205.—*Open Flames.*

(a) In machinery rooms there shall be no fire, arc light, hot surface ignitor or flame open to the atmosphere of the room; any device intended to adequately isolate the fire, arc light, hot surface ignitor or flame shall have been approved by a competent laboratory and the administrative authority. Internal combustion engines with hot surface ignition may be started in the usual manner.

Rule 1206.—*Electrical Equipment.*

(*a*) All current-carrying devices, including bus bars, operating at above six hundred (600) volts shall have all live parts in tight enclosures or be provided with suitable insulating covering.

(*b*) Extensive electrical equipment other than that essential to the ice making or refrigerating plant shall not be permitted in the machinery room.

(*c*) Transformers with primary or secondary connections above six hundred (600) volts shall be in locked vaults which, if adjoining the machinery room, shall be separated therefrom, preferably by unpierced walls, ceilings and floors of six (6) inches of reinforced concrete or its equivalent. All conduit ends and wall openings for conduits entering the vault shall be sealed. (See also the National Electrical Code.)

Rule 1207.—*Gauges, Gauge Glasses.*

(*a*) Every equipment containing over one hundred (100) lb. of ammonia shall be provided with pressure gauges; one for indicating the pressure in condenser or high-pressure side and one for indicating the pressure in evaporator or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic-closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots in two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

Rule 1208.—*Exits, Ventilation.*

(*a*) Each machinery room shall have a direct exit to the open air or to a room or hall from which gases from machinery room can be excluded by a close-fitting self-closing door.

(*b*) Vertical and horizontal openings that permit the passage of gases to other parts of the building, which are not a part of the refrigerating plant, shall be sealed or provided with close-fitting self-closing doors.

(*c*) Each machinery room shall be provided with means for adequate ventilation to the open air, either direct or by means of a suitable duct or ducts. When a mechanical system or ventilation is employed, the system shall be capable of producing a complete change of air at least once in five (5) minutes. The control of such mechanical system shall be easily accessible and located outside of the machinery room.

Rule 1209.—*Charging.*

Every equipment shall be charged through a connection located on the low-pressure side.

SECTION 130

Class C Equipment

A Class C Equipment is a refrigerating system containing fifteen (15) lb. of ammonia or less.

Rule 1300.—*Pressure-Limiting Devices*[3] (*Operating Safety Provisions*).

(*a*) Every Class C Equipment containing more than ten (10) pounds of ammonia and that may be subjected to a pressure in excess of the maximum allowable working

[3] It is recommended that only such safety devices which have been designated as suitable for the purpose by a competent disinterested laboratory be used.

pressure, by the action of the pressure-imposing element, shall be provided with a pressure-limiting device to stop the action of the pressure-imposing element.

(*b*) Such pressure-limiting device shall be so constructed and connected as to act directly responsive to the discharge pressure. There shall be no valve, except necessary service valves, between this device and the pressure-imposing element.

(*c*) Pressure-limiting devices shall be set, marked and sealed to stop the action of the pressure-imposing element at a pressure not in excess of the maximum allowable working pressure, or shall be so designed that they cannot be adjusted to operate at a greater pressure than the pressure stamped thereon.

Rule 1302.—*Electrical Equipment.*

(*a*) All electrical equipment shall be installed in accordance with the National Electrical Code.

Part II—Carbon Dioxide Systems

SECTION 200

General Requirements for Carbon Dioxide Systems

(*a*) Every part of the high-pressure side of the system shall be designed for a maximum allowable working pressure of at least twelve hundred (1200) pounds per square inch gauge pressure and every part of the low-pressure side shall be designed for a maximum allowable working pressure of at least six hundred and fifty (650) pounds per square inch gauge pressure. A factor of safety of five (5) shall be used in all designs, except for gauges and control mechanism.

(*b*) Every vessel in the system, except connecting piping, shall be marked with the maximum allowable working pressure and shall be tested hydrostatically to one and one-half ($1\frac{1}{2}$) times this pressure marking, such tests to be conducted in accordance with the "Unfired Pressure Vessel Code."

SECTION 210

Class A Equipment

A "Class A Equipment" is a refrigerating system of thirty (30) tons' capacity or over, or containing one thousand (1000) pounds of carbon dioxide or over.

Rule 2100.—*Pressure-Imposing Element.*

(*a*) Every equipment shall be provided with one pressure relief valve of proper size connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressures, during the period of excess pressure, and discharge into the machinery room. These pressure relief valves shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure. The proper (nominal pipe) sizes of the pressure relief valves shall be as follows:

For equipment of from	30 ton to 175 ton, use at least one ½ in. valve.
	175 ton to 450 ton, use at least one ¾ in. valve.
	450 ton to 900 ton, use at least one 1 in. valve.

(*b*) Every equipment shall have an automatic pressure-limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the pressure relief valve.

(*c*) A cushioned or other satisfactory check valve shall be placed in the discharge line between the pressure-imposing element and the condenser.

Rule 2101.—*Liquid Receiver, Condenser, and Shell Type Evaporator.*

(*a*) Each liquid receiver, battery of double pipe condenser, shell type condenser, and shell type evaporator which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch pressure relief valve set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point to discharge into the machinery room.

(*b*) The low-pressure side of every equipment, except where the cooling surface is entirely submerged in a liquid, shall be provided with a one-half ($\frac{1}{2}$) inch pressure relief valve designed to prevent pressures in the low-pressure side exceeding the maximum allowable working pressure, and discharging into the machinery room.

(*c*) In place of each relief device (Rule 2101 (*a*) and(*b*)), a rupture member may be employed and its discharge side piped to the outside of the building. The rupture member shall be designed to rupture at the maximum allowable working pressure of the vessel to which it is connected.

(*d*) Piping from rupture members may be joined to a common pipe and thence piped to the outside of the building. No piping from rupture members shall be less than one-half ($\frac{1}{2}$) inch iron pipe size, and for each one hundred (100) feet of run the pipe shall increase one pipe size. The free end of any pipe shall be protected by a suitable gooseneck or screen.

(*e*) The rupture members shall be designed to prevent their rupture parts entering the discharge pipe connections.

(*f*) The rupture orifice for any rupture member shall be not less than one-quarter ($\frac{1}{4}$) inch in diameter and the orifice diameter shall be legibly stamped on the outside of the rupture member body.

(*g*) In place of pressure relief valves mentioned in Rules 2100 (*a*) and 2101 (*a*), (*b*), (*c*), and (*d*) combination devices consisting of a rupture element and a relief valve may be used. These devices shall function at the maximum allowable working pressure. The dimensions of the combination devices shall be the same as specified for relief valves and rupture members, except that no rupture element in a combination device shall have an orifice less than one-half ($\frac{1}{2}$) inch in diameter. The relief element may discharge into the machinery room. This combination device shall be so designed that pressure is exerted against the relief valve only after the rupture element has functioned.

Rule 2102.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser, or high-pressure side, and one for indicating pressure in evaporator or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

Rule 2103.—*Exits, Ventilation.*

(*a*) Each machinery room shall have a direct exit to the open air, or to a room or hall from which gases from machinery room can be excluded by close fitting door.

(*b*) Each machinery room shall be provided with adequate ventilation, as follows:

(1) Where the machinery room floor level is not more than four (4) feet below grade, natural ventilation, to outside atmosphere, by means of doors or windows or both may be employed.

(2) Where the combined floor area of the machinery room and adjacent halls or rooms is not less than one (1.0) square foot for each pound of carbon dioxide in the system, no mechanical means of ventilation need be employed, provided:

a′. The machinery room and adjacent halls or rooms are connected with the machinery room by means of at least two doors or windows or equivalent wall openings and the sills of such doors, windows or equivalent wall openings are not more than twelve (12) inches above the machinery room floor.

b′. The floor level of the adjacent halls or rooms is not more than three (3) feet below the sill of a connecting door, window or equivalent wall opening.

(3) Where a mechanical system of ventilation is employed, the control of such system shall be easily accessible and located outside of the machinery room. The center line of the fan suction or the intake ends of the suction duct shall be within twelve (12) inches of the machinery room floor level.

Rule 2104.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low pressure side.

SECTION 220

Class B Equipment

A "Class B Equipment" is a refrigerating system of less than 30-ton capacity, or containing less than one thousand (1000) pounds of carbon dioxide and over twenty-five (25) pounds of carbon dioxide.

Rule 2200.—*Pressure-Imposing Element.*

(*a*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall be provided with one pressure relief valve of at least one-half (½) inch (nominal pipe) size connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressures, during the period of excess pressure, and discharge into the machinery room. This pressure relief valve shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure.

(*b*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall have an automatic pressure limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the relief valve.

Rule 2201.—*Liquid Receiver, Condenser and Shell Type Evaporator.*

(*a*) Each liquid receiver, battery of double pipe condenser, shell type condenser and shell type evaporator, which can be isolated, shall each be equipped with a one-half ($\frac{1}{2}$) inch pressure relief valve set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point to discharge into the machinery room.

(*b*) The low-pressure side of every equipment containing more than five hundred (500) pounds of carbon dioxide, except where the low-pressure side cooling surface is entirely submerged in a liquid, shall be equipped with not less than a one-half ($\frac{1}{2}$) inch pressure relief valve set to discharge at a pressure not higher than the maximum allowable working pressure.

(*c*) On equipments containing less than five hundred (500) pounds of carbon dioxide, a rupture member of not less than one-quarter ($\frac{1}{4}$) inch diameter of orifice may be provided as an alternate to the pressure relief valve. The rupture member shall be designed to rupture at the maximum allowable working pressure, and may be fitted to discharge into the machinery room.

(*d*) In place of each pressure relief valve (Rule 2201 (*a*) and (*b*)), a rupture member may be employed and its discharge side piped to the outside of the building, except that in equipments containing less than five hundred (500) pounds of carbon dioxide, the rupture member may discharge into the machinery room. The rupture member shall be designed to rupture at the maximum allowable working pressure of the vessel to which it is connected.

(*e*) Piping from rupture members may bejoined to a common pipe and thence piped to the outside of the building. No piping from rupture members shall be less than one-half ($\frac{1}{2}$) inch iron pipe size and for each one hundred (100) feet of run the pipe shall increase one pipe size. The free end of any pipe shall be protected by a suitable gooseneck or screen.

(*f*) The rupture members shall be designed to prevent their ruptured parts entering the discharge pipe connections.

(*g*) The rupture orifice for any rupture member shall be not less than one-quarter ($\frac{1}{4}$) inch in diameter and the orifice diameter shall be legibly stamped on the outside of the rupture member body.

(*h*) In place of pressure relief devices mentioned in Rules 2200 (*a*) and 2201 (*a*), (*b*), (*c*), and (*d*), combination devices consisting of a rupture element and a pressure relief valve may be used. These devices shall function at the maximum allowable working pressure. The dimensions of the combination devices shall be the same as specified for relief valves, except that no rupture element in a combination device shall have an orifice less than one-half ($\frac{1}{2}$) inch in diameter. The relief element may discharge into the machinery room. This combination device shall be so designed that pressure is exerted against the relief valve only after the rupture element has functioned.

Rule 2202.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser or high-pressure side and one for indicating pressure in evaporator or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

Rule 2203.—*Exits, Ventilation.*

(*a*) Each machinery room shall have a direct exit to the open air, or to a room or hall from which gases from machinery room can be excluded by close fitting door.

(*b*) Each machinery room shall be provided with adequate ventilation, as follows:

(1) Where the machinery room floor level is not more than four (4) feet below grade, natural ventilation, to outside atmosphere, by means of doors or windows or both may be employed.

(2) Where the combined floor area of the machinery room and adjacent halls or rooms is not less than one (1.0) square foot for each pound of carbon dioxide in the system, no mechanical means of ventilation need be employed, provided:

a′. The machinery room and adjacent halls or rooms are connected with the machinery room by means of at least two doors or windows or equivalent wall openings and the sills of such doors, windows or equivalent wall openings are not more than twelve (12) inches above the machinery room floor.

b′. The floor level of the adjacent halls or rooms is not more than three (3) feet below the sill of a connecting door, window or equivalent wall opening.

(3) Where a mechanical system of ventilation is employed, the control of such system shall be easily accessible and located outside of the machinery room. The center line of the fan suction or the intake ends of the suction duct shall be within twelve (12) inches of the machinery room floor level.

(*c*) On systems containing less than five hundred (500) pounds of carbon dioxide and where the combined machinery room floor space and adjacent floor space approximately on the same level, communicable by door, is not less than two hundred (200) square feet floor area, no mechanical means of ventilation need be provided.

Rule 2204.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low-pressure side.

SECTION 230

Class C Equipment

A "Class C Equipment" is a refrigerating system containing twenty-five (25) pounds of carbon dioxide or less.

Rule 2300.—*Pressure Limiting Devices*[4] (*Operating Safety Provisions*).

(*a*) Every Class C Equipment shall be provided with at least one rupture member designed to rupture at not higher than the maximum allowable working pressure. The rupture member may discharge into the building space occupied by the equipment.

[4] It is recommended that only such devices which have been designated as suitable for the purpose by a competent disinterested laboratory be used.

(*b*) Every Class C Equipment containing more than fifteen (15) pounds of carbon dioxide and that may be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall be provided with a pressure-limiting device to stop the action of the pressure-imposing element.

(*c*) Such pressure-limiting device shall be so constructed and connected as to act directly responsive to the discharge pressure. There shall be no valve, except necessary service valves, between this device and the pressure-imposing element.

(*d*) Pressure-limiting devices shall be set, marked and sealed to stop the action of the pressure-imposing element at a pressure not in excess of the maximum allowable working pressure, or shall be so designed that they cannot be adjusted to operate at a greater pressure than the pressure stamped thereon.

Rule 2301.—*Electrical Equipment.*

(*a*) All electrical equipment shall be installed in accordance with the National Electrical Code.

Part III—Sulphur Dioxide Systems

SECTION 300

General Requirements for Sulphur Dioxide Systems

(*a*) Every part of the high-pressure side of the system shall be designed for a working pressure of at least seventy-five (75) pounds per square inch gauge pressure and every part of the low-pressure side shall be designed for a working pressure of at least thirty-five (35) pounds per square inch gauge pressure. A factor of safety of five (5) shall be used in all designs, except for gauges and control mechanism.

(*b*) Every vessel in the system, except connecting piping, shall be marked with the maximum allowable working pressure and shall be tested hydrostatically to one and one-half ($1\frac{1}{2}$) times this pressure marking, such tests to be conducted in accordance with the "Unfired Pressure Vessel Code."

SECTION 320

Class B Equipment

A "Class B Equipment" is a refrigerating system of less than thirty (30) tons' capacity, or containing less than one thousand (1000) pounds and over fifteen (15) pounds of sulphur dioxide.

Rule 3200.—*Pressure-Imposing Element.*

(*a*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall be provided with a one-half ($\frac{1}{2}$) inch automatic by-pass connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressure into the low-pressure side on either side of the main suction stop valve; or, as an alternative, through a one-half ($\frac{1}{2}$) inch (standard pipe) pressure relief valve discharging as specified in Rule 3203. These automatic by-passes or pressure relief valves shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure.

(*b*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall have an automatic pressure-limiting device, acting directly

responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the maximum allowable working pressure.

Rule 3201.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch automatic by-pass, set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point, to discharge to that part of the low-pressure side of the equipment protected by an emergency relief device. As an alternative, a pressure relief valve discharging to the atmosphere may be used. (See Rule 3203.)

Rule 3202.—*Fire Emergency Devices.*

(*a*) Every equipment shall be provided with a one-half ($\frac{1}{2}$) inch pressure relief valve connected to the low-pressure side of the equipment, set to relieve excessive pressures in a manner specified in Rule 3203. These devices shall discharge at a pressure not higher than the maximum allowable working pressure.

(*b*) Class B Equipment containing less than one hundred (100) pounds of sulphur dioxide and so designed or provided with devices as to prevent the existence of pressures in any part of the equipment in excess of the maximum allowable working pressure, may be protected by one pressure relief valve.

(*c*) Class B Equipment containing less than one hundred (100) pounds of sulphur dioxide, in which the design of the pressure-imposing element is such that the generation of pressure in excess of the maximum allowable working pressure is impossible, may be protected by fusible plugs on each part of equipment containing sulphur dioxide which can be exposed to heat from an external source, provided the pressure generated at the temperature necessary to fuse the plug is less than the maximum allowable working pressure.

Rule 3203.—*Refrigerant Discharge.*

(*a*) The sulphur dioxide passed by the pressure relief devices shall be discharged either to the atmosphere or to a suitable absorbent.

(*b*) When discharged into the atmosphere, the sulphur dioxide shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the sulphur dioxide with air. The piping from the pressure relief device to point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the device. Such piping shall be used only for sulphur dioxide discharge.

Rule 3204.—*Helmets or Masks.*

(*a*) At least one (1) helmet or mask of approved make, tested in accordance with the requirements of the United States Bureau of Mines for sulphur dioxide, shall be kept in operative condition and available for the immediate use of responsible attendants, who shall be trained in their use, when the refrigerant charge exceeds twenty-five (25) pounds.

(*b*) Helmets or masks shall be kept in a place easily accessible from outside the machinery room.

Rule 3205.—*Gauges, Gauge Glasses.*

(*a*) Every equipment containing over one hundred (100) pounds of sulphur dioxide shall be provided with pressure gauges; one for indicating pressure in con-

denser or high-pressure side and one for indicating pressure in evaporator or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots in two opposites sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

Rule 3206.—*Exits, Ventilation.*

(*a*) Each machinery room shall have a direct exit to the open air or to a room or hall from which gases from machinery room can be excluded by a close fitting, self-closing door.

(*b*) Vertical and horizontal openings that permit the passage of gases to other parts of the building, which are not a part of the refrigerating plant, shall be sealed or provided with close-fitting, self-closing doors.

(*c*) Each machinery room shall be provided with means for adequate ventilation to the open air, either direct or by means of a suitable duct or ducts. Where a mechanical system of ventilation is employed, control of such system shall be located outside the machinery room, where it can be quickly and safely reached and the system started in case of an emergency. The mechanical system shall be of sufficient capacity to change the air in the machinery room at least once every five (5) minutes.

Rule 3207.—*Charging.*

Every equipment shall be charged through a connection located on the low-pressure side.

SECTION 330

Class C Equipment

A Class C Equipment is a refrigerating system containing fifteen (15) pounds of sulphur dioxide or less.

Rule 3300.—*Pressure-Limiting Devices*[5] (*Operating Safety Provisions*).

(*a*) Every Class C Equipment containing more than (10) pounds of sulphur dioxide and that may be subjected to a pressure in excess of the maximum allowable working pressure, by the action of the pressure-imposing element, shall be provided with a pressure-limiting device to stop the action of the pressure-imposing element.

(*b*) Such pressure-limiting device shall be so constructed and connected as to act directly responsive to the discharge pressure. There shall be no valve, except necessary service valves, between this device and the pressure-imposing element.

(*c*) Pressure-limiting devices shall be set, marked and sealed to stop the action of the pressure-imposing element at a pressure not in excess of the maximum allowable working pressure, or shall be so designed that they cannot be adjusted to operate at a greater pressure than the pressure stamped thereon.

Rule 3301.—*Electrical Equipment.*

(*a*) All electrical equipment shall be installed in accordance with the National Electrical Code.

[5] It is recommended that only such devices which have been designated as suitable for the purpose by a competent disinterested laboratory be used.

Part IV—Ethyl and Methyl Chloride Systems

SECTION 400

General Requirements for Ethyl and Methyl Chloride Systems

(*a*) Every part of the high-pressure side of the system shall be designed for the following working pressures:

Systems using { Ethyl Chloride, at least 35 lb. per sq. in gauge pressure.
Methyl Chloride, at least 110 lb. per sq. in. gauge pressure.

Every low-pressure side shall be designed for the following working pressure as follows:

Systems using { Ethyl Chloride, at least 25 lb. per sq. in. gauge pressure.
Methyl Chloride, at least 80 lb. per sq. in. gauge pressure.

A factor of safety of five (5) shall be used in all designs, except for gauges and control mechanism.

(*b*) Every vessel in a refrigerating system, except connecting piping, shall be marked with the maximum allowable working pressure and shall be tested hydrostatically to one and one-half ($1\frac{1}{2}$) times this pressure marking. Such tests to be conducted in accordance with the "Unfired Pressure Vessel Code."

SECTION 420

Class B Equipment

A "Class B Equipment" is a refrigerating system of less than thirty (30) tons' capcity, or containing less than one thousand (1000) pounds and over fifteen (15) pounds of refrigerant.

Rule 4200.—*Pressure-Imposing Element.*

(*a*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall be provided with a one-half ($\frac{1}{2}$) inch automatic by-pass connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressure into the low-pressure side on either side of the main suction stop valve; or, as an alternative, through a one-half ($\frac{1}{2}$) inch (standard pipe) pressure relief valve discharging as specified in Rule 4203 (*a*). These pressure relief devices shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure.

(*b*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall have an automatic pressure-limiting device, acting directly responsible to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the relief valve.

Rule 4201.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one half ($\frac{1}{2}$) automatic by-pass, set to discharge at a pressure not higher than the maximum allowable working pressure and connected at the highest point, to discharge to the low-pressure side of the equip-

ment. As an alternative, a pressure relief device discharging to the atmosphere may be used. See Rule 4203 (*a*).

Rule 4202.—*Fire Emergency Devices.*

(*a*) Every equipment shall be provided with a one-half ($\frac{1}{2}$) inch pressure relief valve connected to the low-pressure side of the equipment, set to relieve excessive pressures in a manner specified in Rule 4203 (*a*), and shall discharge at a pressure not higher than the maximum allowable working pressure.

(*b*) Class B Equipment containing less than 100 pounds of refrigerant and so designed or provided with devices as to prevent the existence of pressures in any part of the equipment in excess of the maximum allowable working pressure may be protected by one pressure relief valve.

Rule 4203.—Refrigerant Discharge.

(*a*) When discharged into the atmosphere, the refrigerant shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the refrigerant with air. The piping from the pressure relief valve to point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the valve. Such piping shall be used only for refrigerant discharged and be so located that the gases discharged are not liable to become ignited. Outlets from diffusers shall be above the roof of any building within fifty (50) feet.

(*b*) Outlets from atmospheric pressure relief devices shall connect to the discharge pipe specified in Paragraph (*a*) of this rule, or a similar pipe.

(*c*) The fire department, if any, shall make at least an annual inspection of the equipment.

Rule 4204.—*Open Flames.*

(*a*) In machinery rooms there shall be no fire, arc light, or flame open to the atmosphere of the room; any device intended to adequately isolate the fire, arc light, or flame shall have been approved by a competent laboratory, and the administrative authority.

Rule 4205.—*Electrical Equipment.*

(*a*) All current carrying devices, bus bars, commutators, collector rings, brushes, etc., shall have all live parts in tight enclosures, or be provided with suitable insulating covering.

(*b*) Extensive electrical equipment other than that essential to the ice making or refrigerating plant shall not be permitted in the machinery room.

(*c*) All starting equipment, if placed in the machinery room, including switches, automatic starters, etc., shall be of the oil immersed or vapor-proof type.

(*d*) If standard open type starting equipment is used, it shall be installed in a room separated from the machinery room by vapor-tight incombustible partition. There shall be no door leading directly from this room to the machinery room. Stationary fireproof windows may be installed. A remote control switch or switches of approved vapor-proof type may be installed in the machinery room for operating the starters, etc., located in the switchboard room.

(*e*) Transformers shall be in locked vaults, which, if adjoining the machinery room, shall be separated therefrom preferably by unpierced walls, ceilings and floors of six (6) inches of reinforced concrete or its equivalent. All conduit ends and wall openings for conduits entering the vault shall be sealed. See also, the National Electrical Code.

Rule 4206.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser, or high-pressure side, and one for indicating pressure in evaporator, or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

Rule 4207.—*Exits, Ventilation.*

(*a*) Each machinery room shall have direct exit to the open air.

(*b*) The machinery room shall be independently provided with means for adequate ventilation to the open air.

(*c*) When a mechanical system of ventilation is employed, the system shall be capable of producing a complete change of air at least once in five (5) minutes. The control of such mechanical system shall be easily accessible and located outside of the machinery room.

(*d*) When windows are used, such windows shall have an area equal to at least one square foot to each five (5) square feet of floor area.

Rule 4208.—*Location.*

(*a*) Machines containing between 12 and 100 pounds of refrigerant shall not be installed in any building in congested districts used for any other than refrigerating purposes, unless such building is of fireproof construction, or all parts of the equipment containing the refrigerant are isolated by fireproof enclosures.

(*b*) Machines containing from 100 to 1000 pounds of refrigerant located in congested districts, shall be installed only in fireproof buildings of not more than one story in height, and shall be located on the grade floor, which shall be of unpierced fireproof construction. The machinery room shall be cut off from the rest of the building by unpierced fireproof walls of not less than eight inches of brick or six inches of reinforced concrete. A direct exit to the open air shall be provided. All the refrigerant shall be confined to the machinery room.

Rule 4209.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low-pressure side.

Rule 4210.—*Precautions.*

(*a*) "No Smoking" regulations shall be vigorously enforced.

SECTION 430

Class C Equipment

A "Class C Equipment" is a refrigerating system containing fifteen (15) pounds of refrigerant or less.

Rule 4300.—*Pressure Limiting Devices*[6] (*Operating Safety Provisions*).

(*a*) Every Class C Equipment containing more than ten (10) pounds of refrigerant that may be subjected to a pressure in excess of the maximum allowable working

[6] It is recommended that only such devices which have been designated as suitable for the purpose by a competent disinterested laboratory be used.

pressure, by the action of the pressure imposing element, shall be provided with a pressure limiting device to stop the action of the pressure imposing element.

(*b*) Such pressure limiting device shall be so constructed and connected as to act directly responsive to the discharge pressure. There shall be no valve, except necessary service valves, between this device and the pressure imposing element.

(*c*) Pressure limiting devices shall be set, marked and sealed to stop the action of the pressure imposing element at a pressure not in excess of the working pressure, or shall be so designed that they cannot be adjusted to operate at a greater pressure than the pressure stamped thereon.

Rule 4301.—*Electrical Equipment.*

(*a*) All electrical equipment shall be installed in accordance with the National Electrical Code.

Part V—Hydro-Carbon Systems

SECTION 500

General Requirements for Hydro-carbon Systems

(*a*) Every part of the high-pressure side of the system shall be designed for the following maximum allowable working pressures:

Systems using	Ethane,	at least	1000 lb. per sq. in. gauge pressure.
	Propane,	at least	200 lb. per sq. in. gauge pressure.
	Freezol,	at least	75 lb. per sq. in. gauge pressure.
	Butane,	at least	50 lb. per sq. in. gauge pressure.

Every part of the low-pressure side shall be designed for the following maximum allowable working pressures:

Systems using	Ethane,	at least	500 lb. per sq. in. gauge pressure.
	Propane,	at least	150 lb. per sq. in. gauge pressure.
	Freezol,	at least	35 lb. per sq. in. gauge pressure.
	Butane,	at least	25 lb. per sq. in. gauge pressure.

A factor of safety of five (5) shall be used in all designs, except for gauges and control mechanism.

(*b*) Every vessel in the system, except connecting piping, shall be marked with the maximum allowable working pressure and shall be tested hydrostatically to one and one-half ($1\frac{1}{2}$) times this pressure marking, such tests to be conducted in accordance with the "Unfired Pressure Vessel Code."

SECTION 510

Class A Equipment

ETHANE—PROPANE

A "Class A Equipment" is a refrigerating system of thirty (30) tons capacity or over, or containing one thousand (1000) pounds of refrigerant or over.

Rule 5100.—*Pressure Imposing Element.*

(*a*) Every equipment shall be provided with one or more automatic by-passes of proper size, connected between each main discharge stop valve and the pressure-imposing element, to relieve excessive pressures into the low-pressure side, on either

side of the main suction stop valve; or, as an alternative, shall be provided with one or more pressure relief valves of proper size discharging, during the period of excess pressure, to the outside atmosphere. (See Rule 5103 (*a*).) These pressure relief devices shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure. The proper (nominal pipe) sizes of such devices shall be as follows:

For equipment of from	30 to 60 tons capacity, use at least one $\frac{3}{4}$ in. device.
	60 to 100 tons capacity, use at least one 1 in. device.
	100 to 175 tons capacity, use at least one $1\frac{1}{4}$ in. device.
	175 to 250 tons capacity, use at least one $1\frac{1}{2}$ in. device.
	250 to 450 tons capacity, use at least one 2 in. device.
	450 to 900 tons capacity, use at least two 2 in. devices.

(*b*) Every equipment shall have an automatic pressure-limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than 90% of the pressure stamped on the relief device.

(*c*) A cushioned or other satisfactory check valve shall be placed in the discharge line between the pressure-imposing element and the condenser.

Rule 5101.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch automatic by-pass set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point to discharge to that part of the low-pressure side of the equipment protected by an emergency relief device. As an alternative, a pressure relief valve discharging to the atmosphere may be used. (See Rule 5103 (*a*).)

Rule 5102.—*Fire Emergency Devices.*

(*a*) The low-pressure side of every equipment shall be provided with a hand operated relief valve for discharging the refrigerant in case of fire to the atmosphere.

(*b*) The hand operated relief valve shall be located on the outside of the wall of the engine room, or shall be capable of operation therefrom. The valve or operating mechanism shall have a protecting cover.

(*c*) The minimum (standard pipe) sizes of the hand relief valves shall be as follows:

Where the charge is	1,000 to 1,800 lb., use one $\frac{3}{4}$ in. valve.
	1,800 to 3,000 lb., use one 1 in. valve.
	3,000 to 5,250 lb., use one $1\frac{1}{4}$ in. valve.
	5,250 to 7,500 lb., use one $1\frac{1}{2}$ in. valve.
	7,500 to 13,500 lb., use one 2 in. valve.
	13,500 to 27,000 lb., use one 3 in. valve.

Rule 5103.—*Refrigerant Discharge.*

(*a*) When discharged into the atmosphere, the refrigerant shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the refrigerant with air. The piping from the hand relief valve to point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the valve. Such piping shall be used only for refrigerant discharge and be so located that the gases discharged are not liable to become ignited. Outlets from diffusers shall be above the roof of any building within fifty (50) feet.

(*b*) Outlets from atmospheric pressure relief devices shall connect to the discharge pipe specified in Paragraph (*a*) of this rule, or a similar pipe.

(*c*) The fire department, if any, shall make at least an annual inspection of the equipment.

Rule 5104.—*Open Flames.*

(*a*) In machinery rooms there shall be no fire, arc light or flame open to the atmosphere of the room; any device intended to adequately isolate the fire, arc light or flame shall have been approved by a competent laboratory and the administrative authority.

Rule 5105.—*Electrical Equipment.*

(*a*) All current carrying devices, bus bars, commutators, collector rings, brushes, etc., shall have all live parts in tight enclosures, or be provided with suitable insulating covering.

(*b*) Extensive electrical equipment other than that essential to the ice making or refrigerating plant shall not be permitted in the machinery room.

(*c*) All starting equipment, including switches, automatic starters, etc., if placed in the machinery room, shall be of the oil immersed or vapor-proof type.

(*d*) If standard open type starting equipment is used, it shall be installed in a room separated from the machinery room by a vapor-tight incombustible partition. There shall be no door leading directly from this room to the machinery room. Stationary fireproof windows may be installed. A remote control switch or switches of approved vapor-proof type may be installed in the machinery room for operating the starters, etc., located in the switchboard room.

(*e*) Transformers shall be in locked vaults, which, if adjoining the machinery room, shall be separated therefrom preferably by unpierced walls, ceilings and floors of six (6) inches of reinforced concrete or its equivalent. All conduit ends and wall openings for conduits entering the vault shall be sealed. See also the National Electrical Code.

Rule 5106.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser, or high-pressure side, and one for indicating pressure in evaporator, or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valve. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

Rule 5107.—*Exits, Ventilation.*

(*a*) Each machinery room shall have direct exit to the open air.

(*b*) The machinery room shall be independently provided with means for adequate ventilation to the open air.

(*c*) When a mechanical system of ventilation is employed, the system shall be capable of producing a complete change of air at least once in five (5) minutes. The control of such mechanical system shall be easily accessible and located outside of the machinery room.

(*d*) When windows are used, such windows shall have an area equal to at least one square foot to each five (5) square feet of floor area.

Rule 5108.—*Location.*

(*a*) Machines shall not be installed in any buildings located in congested districts.

(*b*) Direct expansion shall not be used, unless all parts of the equipment containing refrigerant be confined within fireproof enclosures.

Rule 5109.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low pressure side.

Rule 5110.—*Precautions.*

(*a*) To facilitate the location of leaks, it is recommended that the refrigerant be slightly odorized.

(*b*) "No Smoking" regulations shall be vigorously enforced.

SECTION 520

Class B Equipment

ETHANE—PROPANE—FREEZOL—BUTANE

A "Class B Equipment" is a refrigerating system of less than thirty (30) tons capacity, or containing less than one thousand (1000) pounds and over twelve (12) pounds of refrigerant.

Rule 5200.—*Pressure-Imposing Element.*

(*a*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure imposing element, shall be provided with a one-half ($\frac{1}{2}$) inch automatic by-pass connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressure into the low-pressure side on either side of the main suction stop valve; or, as an alternative, through a one-half ($\frac{1}{2}$) inch (standard pipe) pressure relief valve discharging as specified in Rule 5203 (*a*). These pressure relief devices shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure.

(*b*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall have an automatic pressure-limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the relief device.

Rule 5201.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch automatic by-pass, set to discharge at a pressure not higher than the maximum allowable working pressure and connected at the highest point, to discharge to the low-pressure side of the equipment. As an alternative, a pressure relief valve discharging to the atmosphere may be used. See Rule 5203 (*a*).

Rule 5202.—*Fire Emergency Devices.*

(*a*) Every equipment shall be provided with a one-half ($\frac{1}{2}$) inch pressure relief valve connected to the low-pressure side of the equipment, set to relieve excessive

pressures in a manner specified in Rule 5203 (*a*), and shall discharge at a pressure not higher than the maximum allowable working pressure.

(*b*) Class B Equipment containing less than 100 pounds of refrigerant and so designed or provided with devices as to prevent the existence of pressures in any part of the equipment in excess of the maximum allowable working pressure may be protected by one pressure relief valve.

Rule 5203.—*Refrigerant Discharge.*

(*a*) When discharged into the atmosphere, the refrigerant shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the refrigerant with air. The piping from the pressure relief valve to point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the valve. Such piping shall be used only for refrigerant discharge and be so located that the gases discharged are not liable to become ignited. Outlets from diffusers shall be above the roof of any building within fifty (50) feet.

(*b*) Outlets from atmospheric pressure relief devices shall connect to the discharge pipe specified in Paragraph (*a*) of this rule, or a similar pipe.

(*c*) The fire department, if any, shall make at least an annual inspection of the equipment.

Rule 5204.—*Open Flames.*

(*a*) In machinery rooms there shall be no fire, arc light, or flame open to the atmosphere of the room; any device intended to adequately isolate the fire, arc light, or flame shall have been approved by a competent laboratory and the administrative authority.

Rule 5205.—*Electrical Equipment.*

(*a*) All current carrying devices, bus bars, commutators, collector rings, brushes, etc., shall have all live parts in tight enclosures, or be provided with suitable insulating covering.

(*b*) Extensive electrical equipment other than that essential to the ice making or refrigerating plant shall not be permitted in the machinery room.

(*c*) All starting equipment, including switches, automatic starters, etc., if placed in the machinery room, shall be of the oil immersed or vapor-proof type.

(*d*) If standard open type starting equipment is used, it shall be installed in a room separated from the machinery room by a vapor-tight incombustible partition. There shall be no door leading directly from this room to the machinery room. Stationary fireproof windows may be installed. A remote control switch or switches of approved vapor-proof type may be installed in the machinery room for operating the starters, etc., located in the switchboard room.

(*e*) Transformers shall be in locked vaults, which, if adjoining the machinery room, shall be separated therefrom preferably by unpierced walls, ceilings, and floors of six (6) inches of reinforced concrete or its equivalent. All conduit ends and wall openings for conduits entering the vault shall be sealed. See, also, the National Electrical Code.

Rule 5206.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser, or high-pressure side, and one for indicating pressure in evaporator, or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

Rule 5207.—*Exits, Ventilation.*

(*a*) Each machinery room shall have direct exit to the open air.

(*b*) The machinery room shall be independently provided with means for adequate ventilation to the open air.

(*c*) When a mechanical system of ventilation is employed, the system shall be capable of producing a complete change of air at least once in five (5) minutes. The control of such mechanical system shall be easily accessible and located outside of the machinery room.

(*d*) When windows are used, such windows shall have an area equal to at least one square foot to each five (5) square feet of floor area.

Rule 5208.—*Location.*

(*a*) Machines containing between 12 and 100 pounds of refrigerant shall not be installed in any building in congested districts used for any other than refrigerating purposes, unless such building is of fireproof construction, or all parts of the equipment containing the refrigerant are isolated by fireproof enclosures.

(*b*) Machines containing from 100 to 1000 pounds of refrigerant located in congested districts, shall be installed only in fireproof buildings of not more than one story in height, and shall be located on the grade floor, which shall be of unpierced fireproof construction. The machinery room shall be cut off from the rest of the building by unpierced fireproof walls of not less than eight inches of brick or six inches of reinforced concrete. A direct exit to the open air shall be provided. All the refrigerant shall be confined to the machinery room.

Rule 5209.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low-pressure side.

Rule 5210.—*Precautions.*

(*a*) To facilitate the location of leaks, it is recommended that the refrigerant be slightly odorized.

(*b*) "No Smoking" regulations to be vigorously enforced.

SECTION 530

Class C Equipment

BUTANE—ISOBUTANE—PROPANE

A "Class C Equipment" is a refrigerating system containing twelve (12) pounds of refrigerant or less.

Rule 5300.—*Pressure-Limiting Devices*[7] (*Operating Safety Provision*).

(*a*) Every Class C Equipment containing more than eight (8) pounds of refrigerant that may be subjected to a pressure in excess of the maximum allowable working

[7] It is recommended that only such devices which have been designated as suitable for the purpose by a competent disinterested laboratory be used.

pressure, by the action of the pressure-imposing element, shall be provided with a pressure-limiting device to stop the action of the pressure-imposing element.

(*b*) Such pressure-limiting device shall be so constructed and connected as to act directly responsive to the discharge pressure. There shall be no valve, except necessary service valves, between this device and the pressure-imposing element.

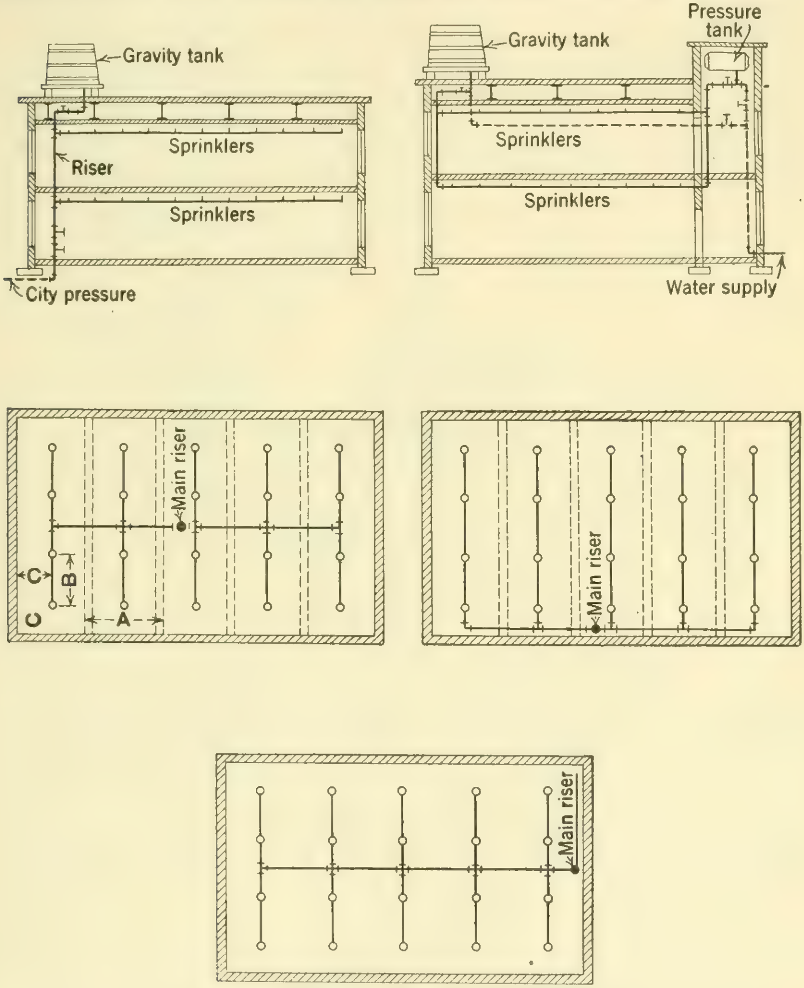


Fig. 382.—Sprinkler Pipe Arrangements.

(*c*) Pressure-limiting devices shall be set, marked and sealed to stop the action of the pressure-imposing element at a pressure not in excess of the working pressure, or shall be so designed that they cannot be adjusted to operate at a greater pressure than the pressure stamped thereon.

Rule 5301.—*Electrical Equipment.*

(*a*) All electrical equipment shall be installed in accordance with the National Electrical Code.

Fire Protection.—Complete fire protection consists of automatic sprinklers, alarm service, inside protection, including a gravity or pressure water tank and an outside protection which must needs be the city water supply (if under sufficient pressure) or a separate fire pump installation.

Sprinklers.—The usual sprinkler head is designed to melt at 165 deg. F., and is designed to be spaced from 8 to 12 ft. apart. For cold storage work it is essential that the piping to the sprinklers be safeguarded against freezing. In such cases a non-freezing (brine) solution can be used or a dry system can be installed using compressed air, in which case (when a sprinkler head melts) the air will pass out through

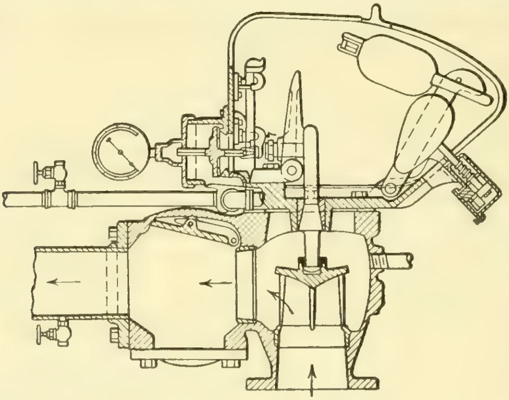


FIG. 383.—Dry Pipe Valve.

the sprinkler head first and will be followed by the water. If a dry pipe sprinkler system is employed, then a calcium chloride reservoir, as shown in Fig. 384, is advised in order to remove the water in the compressed air.

Typical arrangements of piping are shown in Fig. 382. The usual number of sprinklers to be supplied by different sizes of pipe are shown in Table 119.

TABLE 119

Diameter of Pipe, Inches	Maximum Number of Sprinklers	Diameter of Pipe, Inches	Maximum Number of Sprinklers
$\frac{3}{4}$	1	3	36
1	2	$3\frac{1}{2}$	55
$1\frac{1}{4}$	3	4	80
$1\frac{1}{2}$	5	5	140
2	10	6	200
$2\frac{1}{2}$	20		

Fire Pumps.—Every fire protection system should have at least two independent water supplies, one of which must be automatic with a limited water supply and the other must be a connection to the public water works, if under sufficient pressure, or a separate fire pump service

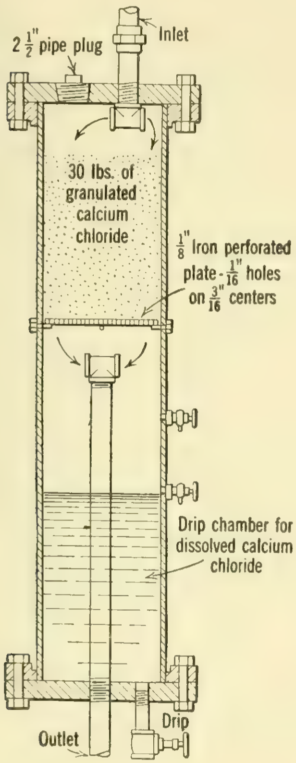


Fig. 384*a*.

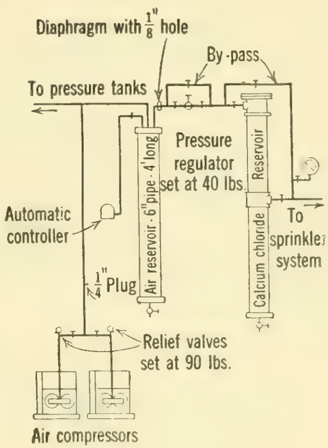


Fig. 384*b*.

Fig. 384.—Calcium Chloride Dryer.

CHAPTER XX

COSTS OF REFRIGERATING MACHINERY AND EQUIPMENT

The subject of *costs* of machinery and equipment can not be handled except in a superficial manner. These values are continually changing with the price of pig iron and steel as well as the economic conditions of supply and demand existing from time to time. These purchase prices increased from 1913 to 1919 and 1920, but have since dropped back, and in 1927 they appear to be practically constant.

However, it will be found that the estimates of various manufacturers will vary over a wide range, for as much as 15 to 20 per cent. In equipment such as the common atmospheric condenser, the double pipe condenser (where the specifications call for a definite amount of pipe and fittings), these prices should be in close agreement, but in the case of compressors, etc., the quality of the materials, the total weight and the machine work and care in the assembly may not be comparable at all.

The following, then, are not intended as absolute values but as approximate only, and are to be used simply as a rough estimate of the possible cost.

TABLE 120

Vertical Enclosed Ammonia Compressors (Belt-driven)

Size, Inches	Tonnage	R.p.m.	Weight, Pounds	Price, Dollars
3 × 3 (single)	$\frac{3}{4}$	255	875	$425
3 × 3 (double)	$1\frac{1}{2}$	258	1,075	510
4 × 4	3	214	1,375	590
5 × 5	5	192	2,050	720
6 × 6	8	170	2,625	880
$6\frac{1}{2}$ × $6\frac{1}{2}$	10	170	2,975	920
$7\frac{1}{2}$ × $7\frac{1}{2}$	15	165	4,300	1200
8 × 8	20	173	5,700	1460
9 × 9	25	155	7,575	1920
10 × 10	35	158	10,800	2930

Above includes gages, bolts and strainer but does not include oil trap.

TABLE 121

APPROXIMATE DATA

"H.D.A. Belt-driven Compressors"

Size, Inches	R.p.m.	B. P., Pounds	C. P., Pounds	Refrigeration, Tons	Weight, Pounds	Price, Dollars
9 ×16	120	19.58	154.5	32.0	12,500	2,400
10½×18	120	19.58	154.5	49.0	19,900	3,300
12 ×24	100	19.58	154.5	71.0	27,100	4,200
13½×26	100	19.58	154.5	97.8	31,900	4.800
"H.D.A. Corliss Engine-driven Compressors"						
9 ×13×26	120	19.58	154.5	32.0	23,000	4,800
10½×15×18	120	19.58	154.5	49.0	29,000	6,700
12 ×18×24	100	19.58	154.5	71.0	45,500	7,100
13½×20×26	100	19.58	154.5	97.8	55,500	8,300
"V.S.A. Belt-driven Compressors"						
9½×12	100	19.46	154.5	23.8	15,500	3,300
12½×18	90	19.46	154.5	55.9	32,500	5,000
13½×20	85	19.46	154.5	68.1	54,000	6,900
15 ×24	80	19.46	154.5	95.3	76,000	8,800
16½×28	75	19.46	154.5	126.3	88,000	10,200
"V.S.A. Corliss Engine-driven Compressors"						
9½×13×12	100	19.46	154.5	23.8	20,400	4,600
12½×18×18	90	19.46	154.5	55.9	37,000	7,000
13½×20×20	85	19.46	154.5	68.1	60,000	11,600
15 ×22×24	80	19.46	154.5	95.3	80,000	15,200
16½×24×28	75	19.46	154.5	126.3	102,000	18,800
"V.S.A. Enclosed Belt-driven Compressors"						
8 × 8	240	19.6	154.5	26.6	7,760	1,400
10 ×10	225	19.6	154.5	48.8	11,660	2,000
12 ×12	200	19.6	154.5	72.0	22,230	3,800
"V.S.A. Semi-enclosed Belt-driven Compressors"						
13 ×14	180	19.0	154.5	95.3	38,500	6,300
15 ×16	164	19.0	154.5	132.5	53,500	8,100
"V.S.A. Enclosed Compressors without Wheel" (for Syn. Motor on Shaft)						
8 × 8	240	19.6	154.5	26.6	5,800	1,300
10 ×10	225	19.6	154.5	48.8	10,000	1,900
12 ×12	200	19.6	154.5	72.0	19,700	3,600
"V.S.A. Semi-enclosed Compressors without Wheel" (for Syn. Motor on Shaft)						
13 ×14	180	19.0	154.5	95.3	37,000	6,100
15 ×16	164	19.0	154.5	132.5	51,000	8,400

TABLE 122

ROLLING MILL TYPE (BELT-DRIVEN)

(Compiled 1920)

Size of Compressor, Inches	Total Weight, Pounds	Selling Price, Dollars	Maximum R.p.m.	Capacity at 20 and 175 Lb.
6×12	7,000	1,320	100	9.65
7×14	8,300	1,460	100	15.32
8×16	10,300	1,670	100	23.65
9×18	13,000	1,940	100	33.61
10×20	18,000	2,420	90	40.40
11×22	22,500	2,870	90	54.0
12×24	24,500	3,295	90	76.2
13×26	32,500	3,790	80	81.5
14×28	36,500	4,210	80	161.6
15×30	42,500	4,795	80	121.8
16×32	48,000	5,570	80	147.0
17×34	52,500	6,300	75	166.2
18×36	68,000	7,730	75	196.8
19×36	75,000	8,610	75	219.8
19×38	78,000	9,030	75	232.0
20×40	88,000	10,300	70	252.0
21×42	104,000	10,870	70	293.5
22×44	....	....	65	312.0
23×46	....	....	65	357.0
24×48	....	....	65	400.0

PRICE OF SUCTION AND DISCHARGE VALVES

Size, Inches	Weight, Pounds	Price, Dollars	Size, Inches	Weight, Pounds	Price, Dollars
$1\frac{7}{8}$	26	30	$1\frac{1}{2}$	26	30
4	65	57	$3\frac{3}{16}$	65	50
6	120	69	$5\frac{1}{4}$	120	69

TABLE 123

Ammonia Receivers

5-ton receiver, 7 in. diameter × 9 ft. long	$ 73
10-ton receiver, 10 in. diameter × 8 ft. long	104
25-ton receiver, 16 in. diameter × 8 ft. long	160
50-ton receiver, 24 in. diameter × 8 ft. long	275
100-ton receiver, 24 in. diameter × 12 ft. long	325

TABLE 124

Ice Making Plants Complete

The mechanical equipment for a 25-ton plant will cost approximately $1100 per ton; for a 50-ton plant, $1000 per ton, and for a 100-ton plant, $900 per ton. These prices would cover machinery, including motors, cranes, tanks, cans, coils, etc., erected in purchaser's building. The approximate prices do not include foundations, insulation, electric wiring, pipe covering and such items, which are generally taken care of by the purchaser and not included in the contract with ice machine manufacturer.

TABLE 125

Horizontal Shell and Tube Brine Coolers

Capacity, Tons	Surface, Square Feet	Diameter, Inches	Length, Feet	Weight, Pounds	Cost, Dollars
4	75	$16\frac{1}{2}$	8	1,500	300
8	150	24	12	3,600	550
25	400	27	16	6,900	950
100	1600	46	16	16,300	200

TABLE 126

STEAM-DRIVEN AMMONIA COMPRESSORS

(Compiled 1920)

Size of Compressor, Inches	Size of Engine, Inches	Frame	Main Bearing, Inches	Weight of Compressor and Engine, Pounds	Price, Total, Dollars
6×12	8×18	00	$4\frac{3}{8}$× 9	14,500	3,040
7×14	8×18	00	$4\frac{3}{8}$× 9	16,000	3,265
8×16	9 ×18	00	$5\frac{7}{8}$× 9	17,500	3,545
9×18	10×24	0	$6\frac{3}{8}$×11	25,500	4,220
10×20	12×24	0	$6\frac{3}{8}$×11	30,500	4,950
11×22	12×30	1	$7\frac{1}{2}$×14	37,000	5,515
12×24	13×30	1	8 ×14	41,500	6,190
13×26	14×30	2	$8\frac{1}{2}$×16	53,000	7,090
14×28	16 ×30	2	9 ×16	58,000	7,990
15×30	16×36	2	$9\frac{1}{2}$×16	64,000	8,440
16×32	18×36	3	10 ×20	74,000	9,230
17×34	20×36	3	11× 20	78,000	9,965
18×36	20×42	4	11 ×24	108,000	12,100
19×36	22×42	4	12 ×24	116,000	12,940
19×38	22×42	4	12 ×24	120,000	13,450
20×40	24×42	4	13 ×24	128,000	14,850
21×42	24×48	5	13 ×28	148,000	16,660
22×44	26×48	5	13 ×28	152,000	17,665
23×46	26×48	5	14 ×28	168,000	19,120
24×48	28×48	5	14 ×28	186,000	21,380

125–150 lb. steam pressure.

TABLE 127

BELT-DRIVEN AMMONIA COMPRESSORS

(Compiled 1920)

Duplex Belt Driven

Size, Inches	Band Wheel			Total Weight, Pounds	Price, Dollars
	Diameter, feet	Face, inches	Weight, pounds		
2– 6×12	7	8	4,500	13,500	2,475
2– 7×14	8	11	5,000	15,500	2,735
2– 8×16	9	15	7,000	19,000	3,400
2– 9×18	9	21	9,500	25,500	3,710
2–10×20	10	26	11,000	34,000	4,500
2–11×22	10	32	14,000	42,000	5,290
2–12×24	14	34	16,500	46,000	5,850

Duplex Rope-driven Heavy Duty

2–13×26	16		14,000	56,000	7,205
2–14×28	18		17,000	64,000	7,950
2–15×30	20		19,500	74,000	9,340
2–16×32	20		23,000	85,000	10,500
2–17×34	22		29,000	94,000	11,770
2–18×36	22		35,000	122,000	13,930

TABLE 128

DOUBLE PIPE AMMONIA CONDENSER

Tons	Length Overall, Inches	Number of Section	Number of Pipes per Section	Total Length of Pipes, Inches	Weight, Pounds	Price, Dollars
3	12	1	6	72	1,200	80
4	20	1	4	80	1,650	75
5	20	1	5	100	1,800	90
6	20	1	6	120	2,000	105
7	20	1	7	140	2,200	135
8	20	1	8	160	2,400	155
10	20	1	10	200	2,700	175
12	20	1	12	240	3,400	230
16	20	2	8	320	4,600	300
18	20	2	9	360	5,600	310
20	20	2	10	400	6,200	345
22	20	2	11	440	6,700	420
25	20	2	12	480	6,900	455
30	20	4	8	560	7,900	600
40	20	4	10	800	11,800	695
50	20	4	12	960	13,500	910
60	20	5	12	1200	15,700	1090
70	20	6	12	1440	19,000	1360
80	20	8	10	1600	23,000	1400
90	20	8	11	1760	24,250	1610
100	20	8	12	1960	25,500	1820
120	20	10	12	2400	27,000	2280
130	20	12	11	2640	29,000	2530
150	20	13	12	3100	36,400	2760

Price includes stands, stop valve on inlet, valves at each end.

TABLE 129

Ammonia Separators

Horizontal NH_3 Separators

Size of Pipes, Inches	For Compressor, Inches		Diameter, Inches	Length, Inches	Thickness of Shell, Inches	Thickness of Heads, Inches	Weight, Pounds	Price, Dollars
	Simple	Duplex						
$1\frac{1}{2}$	6×12	....	8	24	$\frac{9}{32}$	$\frac{3}{8}$	100	50
$2\frac{1}{2}$	9×18	....	10	30	$\frac{5}{16}$	$\frac{1}{2}$	160	75
3	12×24	....	10	30	$\frac{5}{16}$	$\frac{1}{2}$	160	78
4	15×30	12×24	12	30	$\frac{11}{32}$	$\frac{9}{16}$	240	97
5	19×38	....	16	30	$\frac{3}{8}$	$\frac{3}{4}$	350	137
7	23×46	19×36	20	36	$\frac{1}{2}$	$\frac{15}{16}$	700	195
8	....	22×44	20	36	$\frac{1}{2}$	$\frac{15}{16}$	700	230

TABLE 130

Cost of Absorption Machines

Description of Material	25-ton List, Dollars	50-ton List, Dollars	75-ton List, Dollars	100-ton List, Dollars
Generator	2,943	3,878	5,974	6,898
Rectifier (double pipe)	498	836	1,940	1,991
Exchanger shell and coil	1,141	2,294	3,052	4,612
Weak liquor cooler (double pipe)	361	580	1,095	1,672
Condenser (Spiro Flo)	1,380	2,117	3,268	4,234
Absorber, tubular	2,500	4,900	5,900	8,310
Aqua pump	1,300	2,060	2.887	2,887
High and low side connections	1,175	1,960	2,170	2,650
Receivers aqua	238	255	321	402
Receivers anhydrous	200	224	250	320
To insulate generator	140	184	233	270
To insulate exchanger	41	78	89	156
Steam connections	400	600	900	1,200
Water connections	600	900	1,200	1,500
Ammonia charge (anhydrous)	140	215	342	407
Ammonia charge (aqua)	540	950	1,510	1,920
	13,597	22,031	31,131	39,429
Less 50 and 11 per cent to get selling price	6,020	9,800	13,900	17,500

25-ton Absorption Machine Consists of the Following Parts

Generator: 29 in. diameter by 16 ft. 0 in., containing 349 sq. ft. surface in coils. There will be 7 coils made of 2-in. pipe and return bends.

Rectifier: 1 coil, 5 runs made of $1\frac{1}{2}$-in. and 3-in. pipe, of horizontal type, containing 37 sq. ft. of surface.

Exchanger: Of shell and coil type, 1 shell 18 in. diameter by 7 ft. 0 in., containing 187 sq. ft. of surface in the 4 1-in. helical coils.

Weak Liquor Cooler: 1 coil, 5 runs made of 2-in. and 3-in. pipe, containing 60 sq. ft. of surface.

Condenser: 1 vertical tubular spira-flo condenser, $26\frac{1}{2}$ in. diameter by 8 ft. $4\frac{1}{4}$ in., containing 315 sq. ft. surface in the 72 2-in. No. 12 gage seamless steel tubes.

Absorber: 1 tubular absorber, 35 in. diameter by 14 ft. 0 in., containing 706 sq. ft. surface in the 100 2-in. No. 12 gage seamless steel tubes.

Aqua Pump: 1 duplex steam driven aqua pump, 8 by $3\frac{3}{4}$ by 12, capable of delivering 0.573 gal. per stroke.

High and Low Side Connections: Necessary to connect the various pieces of apparatus.

Receiver: 1 12 in. by 4 ft. 0 in. anhydrous and one 24 in. by 4 ft. 0 in. aqua receiver.

Insulation: Necessary magnesia black, plaster and canvas for generator and exchanger shells.

Water Connections: Necessary pipes and fittings for water connections on machine.

Steam and Exhaust Connections: Necessary pipe and fittings for steam and exhaust connections on machine.

Aqua and Anhydrous Charge: Necessary aqua and anhydrous to charge machine.

50-ton Absorption Machine Consists of the Following Parts

Generator: 36 in. diameter by 18 ft. 0 in., containing 654 sq. ft. surface in coils. There will be 9 coils made of 2-in. pipe and return bends.

Rectifier: 1 coil, 6 runs made of 2-in. and 4-in. pipe, of vertical type. Containing 76 sq. ft. surface.

Exchanger: of shell and coil type, 1 shell, 24 in. diameter by 10 ft. 0 in., containing 405 sq. ft. of surface in the 4 $1\frac{1}{4}$-in. helical coils.

Weak Liquor Cooler: 1 coil, 8 runs made of 2-in. and 3-in. pipe, containing 96 sq. ft. of surface.

Condenser: 1 vertical tubular spira-flo condenser, 30 in. diameter by 14 ft. $0\frac{1}{4}$ in., containing 660 sq. ft. of surface in the 90 2-in. No. 12 gage seamless steel tubes.

Absorber: One Tubular absorber 48 in. diameter by 14 ft., containing 1680 sq. ft. surface in the 238 2 in. No. 12 gage seamless steel tubes.

Aqua Pump: One simplex steam driven aqua pump, 12 by 6 by 12, capable of delivering 1.46 gal. per stroke.

High and Low Side Connections: Necessary to connect up various pieces of apparatus.

Receivers: 1 12 in. by 8 ft. 0 in. anhydrous and one 24 in. by 8 in. aqua receiver.

Insulation: Necessary magnesia black, plaster and canvas for generator and exchanger shells.

Water Connections: Necessary pipe and fittings for water connections on machine.

Steam and Exhaust Connections: Necessary pipe and fittings for steam and exhaust connection on machine.

Aqua and Anhydrous Charge: Necessary aqua and anhydrous to charge machine.

75-ton Absorption Machine Consists of the Following Parts

Generator: 45 in. diameter by 18 ft. 0 in., containing 1072 sq. ft. surface in coils. There will be 11 coils made of 2-in. pipe and return bends.

Rectifier: 2 coils, 6 runs each made of 2-in. and 4-in. pipe, of vertical type, containing 122 sq. ft. of surface.

Exchanger: of shell and coil type, 1 shell 29 in. diameter by 9 ft. 0 in., containing 593 sq. ft. surface in the 5 $1\frac{1}{4}$-in. helical coils.

Weak Liquor Cooler: 2 coils, 6 runs each, made of 2-in. and 3-in. pipe, containing 144 sq. ft. surface.

Condenser: 1 vertical tubular spira-flo condenser, consisting of 2 shells, $26\frac{1}{2}$ in. diameter by 13 ft. $1\frac{1}{2}$ in., containing 495 sq. ft. surface in each shell. Each shell has 72 2-in. No. 12 gage seamless steel tubes.

Absorber: 1 tubular absorber, 54 in. diameter by 16 ft. 0 in., containing 2160 sq. ft. of surface in the 268 2-in. No. 12 gage seamless steel tubes.

Aqua Pump: 1 simplex steam driven aqua pump, 14 by 7 by 16, capable of delivering 2.66 gal. per stronke.

High and Low Side Connections: Necessary to connect various pieces of apparatus.

Receivers: 1 12 in. by 12 ft. 0 in. anhydrous and 1 30 in. by 6 ft. 0 in. aqua receiver.

Insulation: Necessary magnesia black, plaster and canvas for generator and exchanger shells.

Water Connections: Necessary pipe and fittings for water connections on machine.

Steam and Exhaust Piping: Necessary pipe and fittings for steam and exhaust connections for machine.

Aqua and Anhydrous: Necessary aqua and anhydrous to charge machine.

100-ton Absorption Machine Consists of the Following Parts

Generator: 45 in. diameter by 21 ft. 0 in., containing 1252 sq. ft. of surface in the coils. There will be 11 coils made of 2-in. pipe return bends.

Rectifier: 2 coils, 6 runs each, made of 2-in. and 4-in. pipe, of vertical type, containing 152 sq. ft. surface.

Exchanger: of shell and coil type, 2 shells 24 in. diameter by 10 ft. 0 in., containing a total of 810 sq. ft. of surface in the 4 $1\frac{1}{4}$-in. helical coils in each shell.

Weak Liquor Cooler: 2 coils, 10 runs each, made of 2-in. and 3-in. pipe, containing 240 sq. ft. surface.

Condenser: 1 vertical tubular spira-flo condenser consisting of 2 shells 30 in. diameter by 14 ft. $0\frac{1}{4}$ in., containing 660 sq. ft. surface in each shell. Each shell will have 90 2-in. No. 12 gage seamless steel tubes.

Absorber: 1 tubular absorber, 63 in. diameter by 16 ft. 0 in., containing 3159 sq. ft. surface in the 392 2-in. No. 12 gage seamless steel tubes.

Aqua Pump: 1 simplex steam driven aqua pump, 14 by 7 by 16, capable of delivering 2.66 gal. per stroke.

High and Low Side Connections: Necessary to connect various pieces of apparatus.

Receiver: 1 18 in. by 7 ft. 6 in. anhydrous and 1 36 in. by 6 ft. 0 in. aqua receiver.

Insulation: Necessary magnesia black, plaster and canvas for generator and exchanger shells.

Water Connections: Necessary pipe and fittings for water connections on machine.

Steam and Exhaust Piping: Necessary pipe and fittings for steam and exhaust connections for machine.

Aqua and Anhydrous: Necessary Aqua and anhydrous to charge machine.

The nearest to a 75-ton cooler we have is one that is rated at 79 tons, having a shell $36\frac{3}{4}$ in. diameter by 13 ft. $1\frac{1}{2}$ in., containing 1025 sq. ft. of surface in the 149 2-in. No. 12 gage seamless steel tube. The selling price of this will be $1255.00.

The nearest to 100-ton cooler we have is one rated at 106 tons, having a shell $46\frac{1}{4}$ in. diameter by 11 ft. $1\frac{1}{2}$ in., containing 1375 sq. ft. surface in the 236 2-in. No. 12 gage seamless steel tubes. Selling price of this will be $1640.00.

These prices are for condensers complete with distributors, cast iron columns, liquid level gage glass, but does not include any valves.

TABLE 131

Coolers and Condensers for CO_2

Horizontal Shell and Tube Brine Cooler

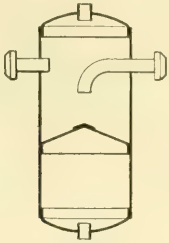
Tons	Diameter		Length		Weight, Pounds	Price, Dollars
	Ft.	In.	Ft.	In.		
8	3	6	13	6	4800	750
10	3	6	13	6	5200	800
12	3	0	20	0	7000	900
16	3	6	20	0	8500	1025

Shell Type Brine Coolers and Condensers—Coil of $\frac{3}{4}$ Ex. Hy. Pipe

3-ton Double Coil Type, 5 ft. 6 in. long, 14 in. diameter, $90
4-ton Double Coil Type, 6 ft. 6 in. long, 14 in. diameter, 130
5-ton Triple Coil Type, 5 ft. 0 in. long, 18 in. diameter, 165
6-ton Triple Coil Type, 6 ft. 0 in. long, 18 in. diameter, 210

TABLE 132

Ammonia Separator or Oil Trap

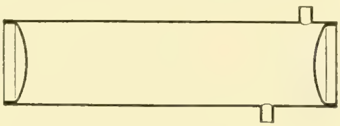


Size of Compressor, Inches	Size of Pipe Connections, Inches	Diameter, Inches	Length, Inches	Weight, Pounds	List Price, Dollars
4×6	1	6	18	70	34.40
5×6	$1\frac{1}{4}$	6	18	75	36.10
6×8	$1\frac{1}{2}$	8	24	90	46.10
7×8	$1\frac{1}{2}$	8	24	100	47.70
8×9 9×9	2	8	24	100	47.70

Price covers standard separator only, as shown, no valves nor companion flanges included. All required valves, companion flanges, fittings, etc. are included in standard high pressure connections.

Horizontal Liquid Receivers

(Compiled 1920)



Size of Compressor, Inches	Diameter, Inches	Length Ft.	Length In.	Inlet and Outlet, Inches	Weight, Pounds	List Price, Dollars
4×6	8	4	6	$\frac{1}{2}$	175	40.00
5×6	8	6	0	$\frac{1}{2}$	200	43.30
6×8	8	10	6	$\frac{3}{4}$	325	53.30
7×8	8	14	0	$\frac{3}{4}$	400	62.20
8×9 9×9	8	14	0	1	400	62.20

Price covers standard receiver only as shown. No valves, companion flanges nor stands are included. All required valves, companion flanges and fittings are included in standard high-pressure connections.

TABLE 133

CO_2 Compressors

Tons	Bore, Inches	Stroke	R.p.m.	Diameter of Wheel, Inches	Weight of Wheel, Pounds	Total Weight, Pounds	Required Horse Power	Price, Dollars
3	$2\frac{1}{4}$	6	120	36	1,000	2,100	$7\frac{1}{2}$	600
4	$2\frac{1}{4}$	6	135	36	1,000	2,100	10	600
5	$2\frac{1}{2}$	9	95	36 and 60	1,700	3,800	10	720
6	$2\frac{3}{4}$	9	110	36 and 60	1,700	3,800	15	720
7	$2\frac{7}{8}$	10	100	36 and 60	2,200	4,300	20	725
8	$2\frac{7}{8}$	10	110	36 and 60	2,200	4,300	20	725
10	3	$11\frac{1}{2}$	90	82	2,600	6,200	20	1115
12	$3\frac{1}{4}$	$11\frac{1}{2}$	90	82	3,000	6,200	25	1115
15	$3\frac{1}{2}$	14	90	82	3,000	9,500	25	1600
18	$3\frac{5}{8}$	14	85	84	4,300	10,500	30	1600
20	$3\frac{3}{4}$	14	95	84	4,500	11,000	35	1600
22	$3\frac{7}{8}$	14	95	84	4,500	11,000	40	1600
25	$4\frac{1}{4}$	16	80	96	6,000	14,500	40	1890
$27\frac{1}{2}$			90					
30	$4\frac{3}{8}$	16	90	96	6,500	15,000	50	1890
40	5	20	70	9	8,000	20,000	60	2290
50	$5\frac{1}{2}$	20	70	9	9,000	21,000	75	2440
60	$5\frac{3}{4}$	20	75	12	10,000	22,000	90	2575
70	$6\frac{1}{4}$	24	85	12	11,500	25,000	100	3320
80	$6\frac{1}{2}$	24	65	14	14,000	27,500	110	3730
90	$6\frac{3}{4}$	24	65	14	15,000	28,500	120	3875
100	7	24	65	14	16,000	29,800	135	4000
120	$7\frac{3}{4}$	28	60	16	18,000	48,000	150	5000
130	8	28	60	18	20,000	49,000	200	3500
150	$8\frac{1}{2}$	28	60	18	25,000	49,000	275	5500

TABLE 134

Triple Pass Shell Type Galvanized Drinking Water Cooler

For CO_2

1 15-ton drinking water cooler, 9 ft. 0 in. long, $22\frac{1}{4}$ in. diameter, weight 4200 lb.	$800
Insulation: 4 vertical partitions with gear cork	100
1 12-ton, 8 ft. 0 in. long by $22\frac{1}{4}$ in. diameter, weight 3800 lb.	725
Insulation	95
1 10 ton, 7 ft. 6 in. long by $22\frac{1}{4}$ in. diameter, weight 3200 lb.	650
Insulation	90
1 8 ton, 6 ft. 0 in. long, $22\frac{1}{4}$ in. diameter, weight 2800 lb.	575
Insulation	85

TABLE 135

DOUBLE PIPE BRINE COOLERS, $1\frac{1}{4}$-IN. AND 2-IN. PIPE, 10 FT. AND 20 FT. LONG OVERALL SPECIFICATION, WEIGHTS AND PRICES

Pipes High	Effective Surface	Effective Surface When Submerged	Height over Top Pipe, Inches	Height Overall, Inches	Full Weight Pipe			Extra Heavy Pipe			Add for Soldered Joints
					Weight, pounds	Black		Weight, pounds	Black		
						Steel	Wrought iron		Steel	Wrought iron	
					10 Ft. Coils						
4	15.0	19.2	24	33	490	$ 98	$112	590	$108	$126	$14
6	19.5	27.8	32	39	710	146	164	840	156	182	20
8	26.0	38.2	40	47	930	190	214	1080	202	236	26
10	32.5	48.0	48	63	1150	230	262	1340	246	288	32
12	39.0	57.6	56	71	1370	266	306	1590	288	338	38
14	45.5	67.2	64	79	1590	298	344	1840	326		
										386	44
16	51.5	77.0	72	87	1810	328	380	2090	362	432	50
18	58.2	86.5	80	95	2030	356	414	2340	396	476	56
20	64.9	96.0	88	113	2250	382	446	2590	428	516	62
					20 Ft. Coils						
4	30.4	44.4	24	33	740	130	158	860	150	196	14
6	45.6	66.6	32	39	1080	186	226	1255	214	262	20
8	61.0	89.0	40	47	1420	236	290	1650	274	342	26
10	76.5	111.0	48	63	1760	286	354	2045	330	414	32
12	91.0	133.2	56	71	2100	330	414	2440	382	482	38
14	106.5	155.5	64	79	2440	370	470	2835	430	546	44

Gas and liquid valves and 2 pr. flanges for brine connections included with each coil. No headers included.

TABLE 136

BRINE ROOM PIPING. BLACK PIPE

Weight and List Price per Foot of Total Pipe in Coil

Length of coil — A =	10 Ft.		20 Ft.		30 Ft.		40 Ft.		60 Ft.		80 Ft.	
	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars
2-in. E. H.-W. I. Lap Weld	5.1	0.98	5.1	0.96	5.1	0.94	5.3	0.92	5.2	0.90	5.2	0.90
2-in. E. H.-Stl. Butt Weld	5.1	.60	5.1	.58	5.1	.56	5.3	.54	5.2	.52	5.2	.52
2-in. F. W.-W. I. Lap Weld	3.7	.78	3.7	.76	3.7	.74	4.0	.72	3.9	.70	3.8	.70
2-in. F. W.-Stl. Butt Weld	3.7	.48	3.7	.46	3.7	.44	4.0	.42	3.9	.40	3.8	.40
$1\frac{1}{4}$-in. E. H.-W. I. Butt Weld	3.2	.56	3.1	.54	3.1	.52	3.5	.52	3.4	.52	3.3	.52
$1\frac{1}{4}$-in. E. H.-Stl. Butt Weld	3.2	.38	3.1	.36	3.1	.34	3.5	.34	3.4	.34	3.3	.34
$1\frac{1}{4}$-in. F. W.-W. I. Butt Weld	2.5	.46	2.3	.44	2.3	.42	2.8	.42	2.7	.42	2.6	.42
$1\frac{1}{4}$-in. F. W.-Stl. Butt Weld	2.5	.30	2.3	.28	2.3	.26	2.8	.26	2.7	.26	2.6	.26
	1.5 ft. and less		2.5 ft.		3.5 ft.		4.5 ft.		5.5 ft.		6.5 ft.	
2-in. E. H.-W. I. Lap Weld	6.0	1.50	5.7	1.26	5.6	1.18	5.5	1.16	5.4	1.14	5.4	1.14
2-in. E. H.-Stl. Butt Weld	6.0	1.16	5.7	.94	5.6	.86	5.5	.84	5.4	.82	5.4	.82
2-in. F. W.-W. I. Lap Weld	......	......	4.4	1.08	4.2	1.02	4.1	1.00	4.0	.98	4.0	.98
2-in. F. W.-Stl. Butt Weld	......	......	4.4	.80	4.2	.74	4.1	.72	4.0	.68	4.0	.68
$1\frac{1}{4}$-in. E. H.-W. I. Butt Weld	4.0	.76	3.6	.74	3.5	.72	3.5	.70	3.4	.66	3.3	.66
$1\frac{1}{4}$-in. E. H.-Stl. Butt Weld	4.0	.60	3.6	.56	3.5	.54	3.5	.52	3.4	.50	3.3	.50
$1\frac{1}{4}$-in. F. W.-W. I. Butt Weld	3.3	.70	3.0	.66	2.8	.64	2.7	.62	2.7	.60	2.6	.60
$1\frac{1}{4}$-in. F. W.-Stl. Butt Weld	3.3	.56	3.0	.52	2.8	.50	2.7	.48	2.7	.46	2.6	.46

No valves, headers, stands nor hangers included.
Coils Nos. 5 and 6 are to be made up with couplings and unions.

TABLE 137

BRINE ROOM PIPING. BLACK PIPE

Weight and List Price per Foot of Total Pipe in Coil

Length of coil — A =	10 Ft.		20 Ft.		30 Ft.		40 Ft.		60 Ft.		80 Ft.	
	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars
2-in. E. H.-W. I. Lap Weld	6.2	1.02	6.1	0.98	5.9	0.96	5.3	0.94	5.4	0.92	5.4	0.90
2-in. E. H.-Stl. Butt Weld	6.2	.64	6.1	.60	5.9	.58	5.3	.56	5.4	.54	5.4	.52
2-in. F. W.-W. I. Butt Weld	4.8	.82	4.7	.78	4.5	.76	3.9	.74	4.0	.72	4.0	.70
2-in. F. W.-Stl. Butt Weld	4.8	.52	4.7	.48	4.5	.46	3.9	.44	4.0	.42	4.0	.40
$1\frac{1}{4}$-in. E. H.-W. I. Butt Weld	4.0	.62	3.9	.60	3.7	.58	3.2	.56	3.3	.54	3.3	.52
$1\frac{1}{4}$-in. E. H.-Stl. Butt Weld	4.0	.44	3.9	.42	3.7	.40	3.2	.38	3.3	.36	3.3	.34
$1\frac{1}{4}$-in. F. W.-W. I. Butt Weld	3.2	.52	3.1	.50	3.0	.48	2.4	.46	2.5	.44	2.5	.42
$1\frac{1}{4}$-in. F. W.-Stl. Butt Weld	3.2	.36	3.1	.34	3.0	.32	2.4	.30	2.5	.28	2.5	.26
	1.5 ft. and less		2.5 ft.		3.5 ft.		4.5 ft.		5.5 ft.		6.5 ft.	
2-in. E. H.-W. I. Lap Weld	6.1	1.02	6.0	1.00	5.8	.96	5.2	.94	5.3	.92	5.3	.90
2-in. E. H.-Stl. Butt Weld	6.1	.64	6.0	.62	5.8	.58	5.2	.56	5.3	.54	5.3	.52
2-in. F. W.-W. I. Lap Weld	4.7	.80	4.6	.78	4.4	.74	3.8	.72	3.9	.70	3.9	.68
2-in. F. W.-Stl. Butt Weld	4.7	.50	4.6	.48	4.4	.44	3.8	.42	3.9	.40	3.9	.38
$1\frac{1}{4}$-in. E. H.-W. I. Butt Weld	3.9	.62	3.8	.60	3.6	.58	3.1	.56	3.2	.54	3.2	.52
$1\frac{1}{4}$-in. E. H.-Stl. Butt Weld	3.9	.42	3.8	.40	3.6	.38	3.1	.36	3.2	.34	3.2	.32
$1\frac{1}{4}$-in. F. W.-W. I. Butt Weld	3.1	.52	3.0	.50	2.9	.48	2.3	.46	2.4	.44	2.4	.42
$1\frac{1}{4}$-in. F. W.-Stl. Butt Weld	3.1	.36	3.0	.34	2.9	.32	2.3	.30	2.4	.28	2.4	.26

Weights and prices of coils Nos. 1 and 3 include stands. No valves, headers nor hangers included.

Coils Nos. 1 and 3 are shipped built up with stands or wood strips bolted on. Coils Nos. 2 and 4 are to be made up with couplings and unions.

TABLE 138

DIRECT EXPANSION ROOM PIPING. BLACK PIPE

Weight and List Price per Foot of Total Pipe in Coil

Length of coil — A =	10 Ft.		20 Ft.		30 Ft.		40 Ft.		60 Ft.		80 Ft.	
	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars
2-in. E. H.-W. I. Lap Weld	5.6	1.10	5.4	0.98	5.3	0.96	5.7	0.96	5.6	0.96	5.6	0.96
2-in. E. H.-Stl. Butt Weld	5.6	.74	5.4	.62	5.3	.60	5.7	.60	5.6	.60	5.6	.60
2-in. F. W.-W. I. Lap Weld	4.2	.90	4.0	.78	4.0	.76	4.3	.76	4.2	.76	4.2	.76
2-in. F. W.-Stl. Butt Weld	4.2	.62	4.0	.50	4.0	.48	4.3	.48	4.2	.48	4.2	.48
$1\frac{1}{4}$-in. E. H.-W. I. Butt Weld	3.6	.66	3.4	.58	3.3	.56	3.5	.56	3.5	.56	3.5	.56
$1\frac{1}{4}$-in. E. H.-Stl. Butt Weld	3.6	.48	3.4	.40	3.3	.38	3.5	.38	3.5	.38	3.5	.38
$1\frac{1}{4}$-in. F. W.-W. I. Butt Weld	2.8	.56	2.6	.48	2.5	.46	2.7	.46	2.7	.46	2.7	.46
$1\frac{1}{4}$-in. F. W.-Stl. Butt Weld	2.8	.42	2.6	.34	2.5	.32	2.7	.32	2.7	.32	2.7	.32
	1.5 ft. and less		2.5 ft.		3.5 ft.		4.5 ft.		5.5 ft.		6.5 ft.	
2-in. E. H.-W. I. Lap Weld	6.0	1.50	5.7	1.26	5.6	1.18	5.5	1.16	5.4	1.14	5.4	1.14
2-in. E. H.-Stl. Butt Weld	6.0	1.20	5.7	.98	5.6	.90	5.5	.88	5.4	.86	5.4	.84
2-in. F. W.-W. I. Lap Weld	......	......	4.4	1.08	4.2	1.02	4.1	1.00	4.0	.98	4.0	.98
2-in. F. W.-Stl. Butt Weld	......	......	4.4	.84	4.2	.78	4.1	.74	4.0	.72	4.0	.72
$1\frac{1}{4}$-in. E. H.-W. I. Butt Weld	4.0	.76	3.6	.74	3.5	.72	3.5	.70	3.4	.66	3.3	.66
$1\frac{1}{4}$-in. E. H.-Stl. Butt Weld	4.0	.62	3.6	.58	3.5	.56	3.5	.54	3.4	.52	3.3	.52
$1\frac{1}{4}$-in. F. W.-W. I. Butt Weld	3.3	.70	3.0	.66	2.8	.64	2.7	.62	2.7	.60	2.6	.60
$1\frac{1}{4}$-in. F. W.-Stl. Butt Weld	3.3	.58	3.0	.54	2.8	.52	2.7	.50	2.7	.48	2.6	.48

No valves, headers, stands nor hangers included. Litharge joints, for soldered joints, see note.
Above prices are for flanges litharged on. For soldering on flanges add per foot as follows:
Coil No. 5, $1\frac{1}{4}$ in.—.02 list per ft. 2 in.—.03 list per ft.
Coil No. 6, $1\frac{1}{4}$ in.—.03 list per ft. 2 in.—.04 list per ft.

TABLE 139

DIRECT EXPANSION ROOM PIPING. BLACK PIPE

Weight and List Price per Foot of Total Pipe in Coil.

Length of coil — A =	10 Ft.		20 Ft.		30 Ft.		40 Ft.		60 Ft.		80 Ft.	
	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars	Weight, pounds	List, dollars
2-in. E. H.-W. I. Lap Weld	6.5	1.08	6.2	1.00	6.0	0.96	5.5	0.98	5.6	0.94	5.6	0.96
2-in. E. H.-Stl. Butt Weld	6.5	.72	6.2	.64	6.0	.60	5.5	.62	5.6	.58	5.6	.60
2-in. F. W.-W. I. Lap Weld	5.2	.88	5.0	.80	4.7	.76	4.2	.78	4.3	.74	4.3	.76
2-in. F. W.-Stl. Butt Weld	5.2	.60	5.0	.52	4.7	.48	4.2	.50	4.3	.46	4.3	.48
$1\frac{1}{4}$-in. E. H.-W. I. Butt Weld	4.2	.66	4.0	.60	3.8	.58	3.2	.58	3.3	.56	3.3	.56
$1\frac{1}{4}$-in. E. H.-Stl. Butt Weld	4.2	.48	4.0	.42	3.8	.40	3.2	.40	3.3	.38	3.3	.38
$1\frac{1}{4}$-in. F. W.-W. I. Butt Weld	3.5	.56	3.3	.50	3.2	.48	2.5	.48	2.6	.46	2.6	.46
$1\frac{1}{4}$-in. F. W.-Stl. Butt Weld	3.5	.40	3.3	.34	3.2	.32	2.5	.32	2.6	.30	2.6	.30
	1.5 ft. and less		2.5 ft.		3.5 ft.		4.5 ft.		5.5 ft.		6.5 ft.	
2-in. E. H.-W. I. Lap Weld	6.4	1.04	6.1	.98	5.9	.96	5.4	.96	5.5	.94	5.5	.96
2-in. E. H.-Stl. Butt Weld	6.4	.68	6.1	.62	5.9	.60	5.4	.60	5.5	.58	5.5	.60
2-in. F. W.-W. I. Lap Weld	5.1	.84	4.9	.78	4.6	.76	4.1	.76	4.2	.74	4.2	.76
2-in. F. W.-Stl. Butt Weld	5.1	.56	4.9	.50	4.6	.48	4.1	.48	4.2	.46	4.2	.48
$1\frac{1}{4}$-in. E. H.-W. I. Butt Weld	4.1	.64	3.9	.58	3.7	.56	3.1	.56	3.2	.54	3.2	.56
$1\frac{1}{4}$-in. E. H.-Stl. Butt Weld	4.1	.46	3.9	.40	3.7	.38	3.1	.38	3.2	.36	3.2	.38
$1\frac{1}{4}$-in. F. W.-W. I. Butt Weld	3.4	.54	3.2	.48	3.1	.46	2.4	.46	2.5	.44	2.5	.46
$1\frac{1}{4}$-in. F. W.-Stl. Butt Weld	3.4	.40	3.2	.34	3.1	.32	2.4	.32	2.5	.30	2.5	.32

Weights and prices of coils Nos. 1 and 3 include stands. Weights and prices of coils Nos. 2 and 4 do not include stands. No valves, headers or hangers included. Litharge joints. For soldered joints, see note.

Above prices are for flanges litharged on. For soldering on flanges add per foot as follows:

Coils Nos. 1 and 3, $1\frac{1}{4}$ in.—.015 list per foot 2 in.—.02 list per foot.

Coils Nos. 2 and 4, $1\frac{1}{4}$ in.—.025 list per foot 2 in.—.03 list per foot.

TABLE 140

PIPE INSULATION (CORK)

	Price
$\frac{1}{2}$-in. pipe	$1.37
$\frac{3}{4}$-in. pipe	1.54
1 -in. pipe	1.60
$1\frac{1}{4}$-in. pipe	1.65
$1\frac{1}{2}$-in. pipe	1.68
2 -in. pipe	1.88
$2\frac{1}{2}$-in. pipe	1.99
3 -in. pipe	2.17

In estimating work of this kind, all fittings to be measured over and counted as an additional foot of pipe.

TABLE 141

STANDARD BELT DRIVE FOR COMPRESSOR

Oak-tanned, Regular Weight, Double-ply Leather

Size of Tons	Total Length of Belt, Feet	Width, Inches	List Price, Dollars	Selling Price, Dollars
3	26	5	62.50	38.00
4	30	5	72.00	44.00
5	30	6	86.50	52.00
6	30	6	86.50	55.00
8	40	8	154.00	95.00
10	40	8	154.00	95.00
12	40	10	192.00	120.00
15	45	10	215.00	130.00
18	45	12	260.00	160.00
20	45	12	260.00	160.00
25	50	14	288.00	175.00
30	50	14	336.00	205.00
40	58	14	380.00	235.00
50	60	14	400.00	245.00
60	60	16	460.00	280.00
70	60	16	460.00	280.00
80	60	16	460.00	280.00
90	80	18	690.00	420.00
100	80	18	690.00	420.00
110	80	18	690.00	420.00
125	100	20	960.00	580.00
150	100	22	1056.00	640.00

10-in. belt for 6-in. vertical 28 ft. long, $80.00.

Waterproof belts are recommended for all installations where there is possibility of dampness.

For waterproof belts add 15 per cent to the above list.

TABLE 142

BRINE STRAINERS

	List Price
$1\frac{1}{4}$ in. and $1\frac{1}{2}$ in.	$4
2 in. and $2\frac{1}{2}$ in.	15
3 in.	22
4 in.	30

TABLE 143

RAW WATER AIR VALVES, HEADERS AND DROP PIPES

Size of Ice Can:	List Price
$6 \times 12 \times 29$, 50 lb.	$2.70 per can
$8 \times 16 \times 32$, 100 lb.	2.95 per can
$11\frac{1}{2} \times 22\frac{1}{2} \times 32$, 200 lb.	3.10 per can
$11\frac{1}{2} \times 22\frac{1}{2} \times 45$, 300 lb.	3.15 per can
Hand type core sucker	19.50

Prices include air piping from blower.

TABLE 144

HORIZONTAL BELT-DRIVEN BRINE AGITATORS

Cost Prices

9 in., $75 net, F.O.B. Pennsylvania
12 in., 100 net, F.O.B. Pennsylvania
20 in., 150 net, F.O.B. Pennsylvania.

Each complete with one pulley.

TABLE 145

CALCIUM CHLORIDE

Solid (650-lb. Barrels)	Per ton	Granulated (375-lb. barrels)	Per ton
1 ton	$45	1 ton	$50
2 to 3 tons	42	2 to 3 tons	49
4 to 8 tons	40	4 to 5 tons	48
9 to 15 tons	38	6 to 9 tons	46
16 tons and over	36	10 tons and over	45

TABLE 146

PIPING FOR CARBON DIOXIDE

Pipe (Extra Heavy Wrought Steel Pipe) installed, per foot:

3 in	$1.70	1¼ in	$0.90
2½	1.50	1	.80
2	1.20	¾	.70
1½	1.00	½	.60

Pipe Covering and Asphaltum Paint Finish, per Foot:

3 in	$4.85	1¼ in	$2.60
2½	4.20	1	2.25
2	3.55	¾	1.95
1	3.00	½	1.75

Galvanized Brine Pipe installed, per Foot:

4 in	$1.83	1½ in	$0.98
3	1.68	1¼	.88
2½	1.43	1	.78
2	1.18	¾	.68
		½	.58

Galvanized Brine Pipe, Asphaltum Paint and Covering, per Foot:

4 in	$5.98	1½ in	$2.58
3	4.83	1¼	2.23
2½	4.18	1	1.93
2	3.53	¾	1.73

TABLE 147

"National" Wrought Steel Pipe

Discounts F.O.B. Chicago

Size	Random Pipe		Size	Random Pipe	
	Black	Galvanized		Black	Galvanized
$\frac{1}{8}$ in., butt......	37	4	2 in., lap.......	41	28
$\frac{1}{4}$ and $\frac{3}{8}$ in. butt.	43	16	$2\frac{1}{2}$ to 6 in., lap..	51	38
$\frac{1}{2}$ in., butt......	48	33	7 to 8 in., lap...	43	29
$\frac{3}{4}$ in., butt......	52	39	9 and 10 in., lap..	41	27
1 to 3 in., butt..	54	41	11 and 12 in., lap.	40	26
Extra Strong Plain Ends					
$\frac{1}{8}$ in., butt......	25		2 in., lap.......	37	13
$\frac{1}{4}$ and $\frac{3}{8}$ in., butt.	31		$2\frac{1}{2}$ to 4 in., lap..	41	17
$\frac{1}{2}$ in., butt......	37	13	$4\frac{1}{2}$ to 6 in., lap..	40	16
$\frac{3}{4}$ in., butt......	42	18	7 to 8 in., lap...	34	10
1 to $1\frac{1}{2}$ in., butt.	44	20	9 and 10 in., lap.	On application	
2 to 3 in., butt..	45	21	11 and 12 in., lap		
Double Extra Strong Plain Ends					
$\frac{1}{2}$ in., butt......	22	On Application	$2\frac{1}{2}$ to 4 in., lap..	23	On Application
$\frac{3}{4}$ to $1\frac{1}{2}$ in., butt.	25		$4\frac{1}{2}$ to 6 in. lap..	22	
2 and $2\frac{1}{2}$ in. butt	27		7 and 8 in. lap..	16	
2 in. lap.......	21				
Net Price per 100 Ft. Standard Weight F.O.B. Chicago					
$\frac{1}{8}$ in. butt.....	\$3.47	\$5.28	4 in. lap.......	\$53.41	\$67.58
$\frac{1}{4}$ in. butt......	3.4	5.04	$4\frac{1}{2}$ in. lap......	62.23	78.74
$\frac{3}{8}$ in. butt......	3.42	5.04	5 in. lap.......	72.52	91.76
$\frac{1}{2}$ in. butt.....	4.42	5.70	6 in. lap.......	94.08	119.04
$\frac{3}{4}$ in. butt.....	5.52	7.02	7 in. lap.......	135.66	168.98
1 in. butt......	7.82	10.03	8 in. 25 lb. lap..	142.50	177.50
$1\frac{1}{4}$ in. butt....	10.58	13.57	8 in. 28 lb., lap..	164.16	204.48
$1\frac{1}{2}$ in., butt....	12.65	16.23	9 in., lap.......	203.55	251.85
2 in., butt.....	17.02	21.83	10 in., 35 lb., lap.	206.50	255.50
$2\frac{1}{2}$ in., butt....	26.91	34.52	10 in., 41 lb., lap.	243.08	300.76
3 in., butt......	35.19	45.14	12 in., 45 lb., lap.	270.00	333.00
2 in., lap.......	21.83	26.64	12 in., 50 lb., lap.	304.20	375.18
$3\frac{1}{2}$ in., lap......	45.08	57.04			

Weight of 8 in.–10 in.–12 in. must be specified.

Tarred pipe, all sizes, and O.D. pipe 14 to 24 in. in stock.
Prices on application.

TABLE 147*a*

"Reading" Wrought Iron Pipe

Discounts F.O.B. Chicago

Size	Random Pipe Black	Random Pipe Galvanized	Size	Random Pipe Black	Random Pipe Galvanized
$\frac{1}{8}$ in.	On application		2 in., lap	7	+11
$\frac{1}{4}$ and $\frac{3}{8}$ in., butt	+27	+57	$2\frac{1}{2}$ in., lap	10	+ 7
$\frac{1}{2}$ in., butt	6	+16	3 to 6 in., lap	12	+ 5
$\frac{3}{4}$ in., butt	12	+ 7	7 to 12 in., lap	8	+ 9
1 to $1\frac{1}{2}$ in., butt	14	+ 5			
Extra Strong Plain Ends					
$\frac{1}{4}$ and $\frac{3}{8}$ in., butt	+40	On Application	$2\frac{1}{2}$ in., lap	8	On Application
$\frac{1}{2}$ in., butt	Net list		3 to 4 in., lap	8	
$\frac{3}{4}$ in., butt	7		$4\frac{1}{2}$ to 6 in., lap	7	
1 to $1\frac{1}{2}$ in., butt	9		7 and 8 in., lap	+ 2	
2 in., lap	2		9 to 12 in., lap	+ 7	
Double Extra Strong Plain Ends					
$\frac{1}{2}$ in., butt	On application		$2\frac{1}{2}$ to 4 in., lap	On application	
$\frac{3}{4}$ to $1\frac{1}{2}$ in., butt			$4\frac{1}{2}$ to 6 in., lap		
2 in., lap					
Net Price per 100 Ft., Standard Weight, F.O.B. Chicago					
$\frac{1}{4}$ in., butt	$7.62	$9.42	4 in., lap	$95.92	$114.45
$\frac{3}{8}$ in., butt	7.62	9.42	$4\frac{1}{2}$ in., lap	111.76	133.35
$\frac{1}{2}$ in., butt	7.99	9.86	5 in., lap	130.24	155.40
$\frac{3}{4}$ in., butt	10.12	12.31	6 in., lap	168.96	201.60
1 in., butt	14.62	17.85	7 in., lap	218.90	259.42
$1\frac{1}{4}$ in., butt	19.78	24.15	8 in., 25 lb., lap		
$1\frac{1}{2}$ in., butt	23.65	28.88	8 in., 28 lb., lap	264.96	313.92
$1\frac{1}{2}$ in., lap			9 in., lap	317.40	367.05
2 in., lap	34.41	41.07	10 in., 35 lb., lap		
$2\frac{1}{2}$ in., lap	52.65	62.60	10 in., 41 lb., lap	379.04	449.08
3 in., lap	67.32	80.33	12 in., 45 lb., lap		
$3\frac{1}{2}$ in., lap	80.96	96.60	12 in., 50 lb., lap	466.44	552.63

Reading wrought iron O.D. pipe up to 20 in. can be furnished on short notice.

TABLE 147*b*

"Alpha" Brass Pipe and Tubing

(Seamless)

Standard Iron Pipe Size—Net Prices per Foot

Size, Inches	Full Lengths	Cut Lengths	Add for Nickel Plating	Threads Each
$\frac{1}{8}$	.11	.14	.08	.07$\frac{1}{2}$
$\frac{1}{4}$	.18	.23	.08	.07$\frac{1}{2}$
$\frac{3}{8}$	.23	.29	.08	.07$\frac{1}{2}$
$\frac{1}{2}$	.31	.34	.13	.07$\frac{1}{2}$
$\frac{3}{4}$	.39	.49	.16	.07$\frac{1}{2}$
1	.55	.69	.19	.09
1$\frac{1}{4}$	.80	1.00	.20	.10$\frac{1}{2}$
1$\frac{1}{2}$	.95	1.18	.21	.12
2	1.25	1.57	.28	.15
2$\frac{1}{2}$	1.82	2.73	.39	.27$\frac{1}{2}$
3	2.60	3.90	.47	.30

Larger sizes on application.

Size O.D., Inches	Seamless Tubing No. 17G Nickel Plated, 12 Ft. Lengths, Net per Foot	Size Pipe to Cover, Inches	Tubing (Casing) Nickel Plated to Cover Wrought Pipe, Net per Foot
1$\frac{1}{4}$	.46	$\frac{3}{8}$	.25
1$\frac{3}{8}$	.50	$\frac{1}{2}$	.30
1$\frac{1}{2}$	.60	$\frac{3}{4}$	.35
1$\frac{3}{4}$	....	1	.45
2	....	1$\frac{1}{4}$	.55
	No. 20G Rough in 12 Ft. Lengths	1$\frac{1}{2}$	.65
1$\frac{1}{2}$	.35		For cutting add 20 per cent to above.

Threading, all sizes of tubing, .25 net per thread.

TABLE 148

ICE CAN COVERS ONLY

Quantity	Dimensions	Price	Remarks
50	$8\frac{1}{4}\times14\frac{1}{4}$	$0.75	3–$\frac{7}{8}$ in. lumber water proof paper between.
100	$10\frac{1}{4}\times18\frac{1}{4}$	1.25	
200	$13\frac{5}{8}\times24\frac{5}{8}$	2.50	
300	$13\frac{5}{8}\times24\frac{5}{8}$	2.50	

TABLE 149

AUTOMATIC SPRINKLER

Size of Cans, Pounds	Automatic Sprinkling Pumps and Pans Complete		
	Floor space	Pan depth, inches	Price
50	2 ft. 0 in. × 3 ft. 9 in.	4	$70
100	2 ft. 8 in. × 4 ft. 10 in.	5	75
200	3 ft. 0 in. × 5 ft. $2\frac{1}{2}$ in.	5	95
300	3 ft. 0 in. × 6 ft. 0 in.	5	110

TABLE 150

SYNCHRONOUS MOTORS

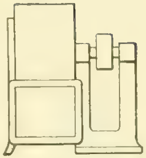
Including motor with base and 2 pedestals for coupling to compressors; also push button across the line, auto-starter and including motor generator exciter sets.

75 hp. 100 per cent P.F. 200 r.p.m., 2 pedestal bearings base and shaft with extension for coupling to compressor across line starter from push button, indicating a.c. ammeter and d.c. ammeter.................... $3150
150 hp., 200 r.p.m., same as above.................................... 3800
225 hp., 200 r.p.m., same as above.................................... 4250
150 hp., 180 r.p.m., same as above.................................... 4160
300 hp., 171 r.p.m., same as above.................................... 5480
450 hp., 164 r.p.m., same as above.................................... 6160

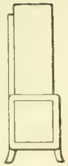
Without shafts or bearings, but including pole plates, split rotor exciter sets and automatic control push button starter:

150 hp., 100 per cent, P.F., 150 r.p.m., 220 volts......................... $4435
150 hp., 100 per cent, P.F., 138 r.p.m., 220 volts......................... 4825
200 hp., 100 per cent, P.F., 150 r.p.m., 220 volts......................... 4640
200 hp., 100 per cent, P.F., 138 r.p.m., 220 volts......................... 4960

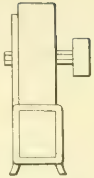
TABLE 151



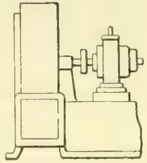
Housing, Pedestal, 2-Bearings, pulley Shaft, Wheel (1)



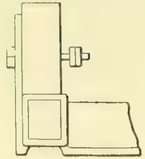
Housing & Wheel only (2)



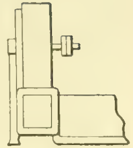
Housing, Wheel & Pedestal for Motor or Engine (3)



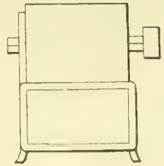
Housing, Wheel & Pedestal for Motor or Engine (4)



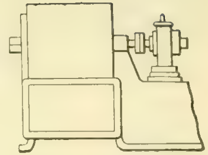
Housing, Wheel Shaft One Intermediate Bearing, Coupling & Pedestal (5)



Housing, Wheel Shaft, One Bearing (in inlet) Pedestal & Coupling (6)



Double Inlet (3)



Double Inlet (7)

Fan Number	Diameter of Wheel, Inches	Single Inlet										Double Inlet			
		Arrangement No. 1		Arrangement No. 2		Arrangement No. 3		Arrangement No. 4		Arrangement Nos. 5 and 6		Arrangement No. 3		Arrangement No. 7	
		Price, dollars	Weight, pounds	Price, dollars	Weight, pounds	Price, dollars	Weight, pounds	Price, dollars	Weight, pounds	Price, dollars	Weight, pounds	Price, dollars	Weight, pounds	Price, dollars	Weight, pounds
00	3	34	10	32	8	......	......	34	10						
0	4½	50	30	48	25	......	......	50	30						
1	6	76	60	70	50	......	......	76	60						
1¼	7½	102	90	96	70	......	......	102	80						
1½	9	134	130	128	100	......	......	134	120						
2	12	168	200	160	170	......	......	168	225						
2½	15	210	320	200	280	......	......	210	370	......	......	354	520	378	550
3	18	270	500	256	400	......	......	270	600	......	......	454	625	488	650
3½	21	354	700	336	500	......	......	354	730	......	......	588	775	630	775
4	24	482	850	414	650	440	620	460	620	482	700	722	1050	772	1130
4½	27	556	1020	480	750	500	800	530	800	556	900	840	1375	890	1500
5	30	628	1350	542	950	570	1050	600	1050	628	1200	950	1950	1010	2150
6	36	776	1800	672	1200	706	1500	740	1500	776	1700	1180	3000	1240	3200

Sirocco fans:	Discount to users	Discount to dealers
Arrangements Nos. 1, 2 and 3	54 per cent	60 per cent
Arrangements Nos. 4, 5, 6 and 7	51 per cent	58 per cent

Add 5 per cent for Troy type. Size 4 and larger

TABLE 152

Type CS Squirrel-Cage Motors

Open Type, 60 Cycles, Continuous Rated 40° C., Constant Speed

Horse-power	Volts	Number of Poles	Full Load, R.p.m.	Frame	Price, Dollars: Complete motor with Type "A" manual starter Class 10,700	Price, Dollars: Complete motor with Magnetic auto-transformer type starter	Price, Dollars: Bare motor with Type "A" manual starter Class 10,700	Price, Dollars: Bare motor with Magnetic auto-transformer type starter	Price, Dollars: Bare motor	Price, Dollars: Omission or addition for Rails	Price, Dollars: Omission or addition for Pulley	Price, Dollars: Addition for vertical parts	Approximate Shipping Weight, Pounds: Complete motor with Type A starter	Approximate Shipping Weight, Pounds: Complete motor with Magnetic starter	Approximate Shipping Weight, Pounds: Bare motor	Approximate Shipping Weight, Pounds: Omission or addition for Rails	Approximate Shipping Weight, Pounds: Omission or addition for Pulley	Pulley Dimensions, Inches: Standard diameter and face	Pulley Dimensions, Inches: Bore	Pulley Dimensions, Inches: Special pulley, minimum diameter and maximum face	Shaft Dimensions, Inches: Standard shaft extension	Shaft Dimensions, Inches: Standard keyway dimensions, L. W. D.
$7\frac{1}{2}$	220, 440, 550	2	3450	243C	.....	.....	193	257	117	6	.....	60	422c	422c	192	30	.....	.....	$1\frac{1}{4}$	.....	$3\frac{11}{16}$	$2\frac{15}{16}\times\frac{15}{16}\times\frac{1}{8}$
		4	1750	250C	190	254	182	246	106	6	2	60	339	439	199	35	5	$5\times4\frac{1}{2}$	$1\frac{1}{4}$	5×5	$3\frac{13}{16}$	$2\frac{15}{16}\times\frac{15}{16}\times\frac{1}{8}$
		6	1160	352C	224	288	210	274	134	11	3	75	443	543	291	41	10	6×5	$1\frac{5}{8}$	$4\frac{1}{2}\times9$	$5\frac{7}{32}$	$4\frac{1}{2}\times\frac{3}{8}\times\frac{1}{8}$
		8	870	370C	259	323	245	309	169	11	3	75	510	610	348	47	15	7×6	$1\frac{5}{8}$	5×10	$5\frac{5}{16}$	$4\frac{1}{2}\times\frac{3}{8}\times\frac{1}{8}$
		10	690	460C	313	377	298	362	222	11	4	80	588	688	425	48	15	8×7	$1\frac{7}{8}$	6×10	$6\frac{11}{16}$	$5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$
		12	575	464C	334	398	319	383	243	11	4	80	625	725	462	48	15	8×7	$1\frac{7}{8}$	6×10	$6\frac{11}{16}$	$5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$
		14	490	566C	380	444	364	428	288	12	4	110	940	1040	735	80	25	9×8	$2\frac{3}{8}$	7×12	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		16	435	586C	392	456	376	440	300	12	4	110	1041	1141	836	80	25	9×8	$2\frac{3}{8}$	9×12	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
10	220, 440, 550	2	3450	351.5C	.....	.....	215	275	135	11	.....	75	398c	498c	256	42	.....	.....	$1\frac{5}{8}$	.....	$4\frac{1}{4}$	$4\frac{1}{2}\times\frac{3}{8}\times\frac{1}{8}$
		4	1750	351C	217	277	203	263	123	11	3	75	408	508	256	42	10	6×5	$1\frac{5}{8}$	4×9	$5\frac{7}{32}$	$4\frac{1}{2}\times\frac{3}{8}\times\frac{1}{8}$
		6	1160	370C	251	311	237	297	157	11	3	75	510	610	348	47	15	7×6	$1\frac{5}{8}$	5×10	$5\frac{5}{16}$	$4\frac{1}{2}\times\frac{3}{8}\times\frac{1}{8}$
		8	870	460C	287	347	272	332	192	11	4	80	588	688	425	48	15	8×7	$1\frac{7}{8}$	6×10	$6\frac{11}{16}$	$5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$
		10	690	480C	350	410	335	395	255	11	4	80	699	799	536	48	15	8×7	$1\frac{7}{8}$	7×10	$6\frac{11}{16}$	$5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$
		12	575	566C	376	436	360	420	280	12	4	110	940	1040	735	80	25	9×8	$2\frac{3}{8}$	7×12	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		14	490	586C	417	477	400	460	320	12	5	110	1056	1156	836	80	40	10×9	$2\frac{3}{8}$	9×12	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		16	435	642C	446	506	425	485	345	16	5	115	1607	1705	1302	165	40	10×9	$2\frac{7}{8}$	9×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
15	220, 440, 550	2	3450	363.5C	.....	.....	251	309	169	11	.....	75	560c	610c	363	47	.....	.....	$1\frac{5}{8}$	.....	$4\frac{1}{4}$	$4\frac{1}{2}\times\frac{3}{8}\times\frac{1}{8}$
		4	1750	371C	252	310	238	296	156	11	3	75	570	620	358	47	15	7×6	$1\frac{5}{8}$	5×10	$5\frac{5}{16}$	$4\frac{1}{2}\times\frac{3}{8}\times\frac{1}{8}$
		6	1160	460C	294	352	279	337	197	11	4	80	638	688	425	48	15	8×7	$1\frac{7}{8}$	6×10	$6\frac{11}{16}$	$5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$
		8	870	480C	330	388	315	373	233	11	4	80	759	809	536	48	25	9×8	$1\frac{7}{8}$	6×10	$6\frac{11}{16}$	$5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$
		10	690	566C	410	468	393	451	311	12	5	110	1005	1055	735	80	40	10×9	$2\frac{3}{8}$	8×12	8	$5\frac{5}{8}\times\frac{9}{16}\times\frac{3}{16}$
		12	575	582C	440	498	423	481	341	12	5	110	1074	1124	804	80	40	10×9	$2\frac{3}{8}$	9×12	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		14	490	642C	495	553	472	530	390	16	7	115	1667	1717	1302	165	50	11×10	$2\frac{7}{8}$	9×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		16	435	644C	525	583	502	560	420	16	7	115	1744	1794	1379	165	50	11×10	$2\frac{7}{8}$	9×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$

20	220 440 550	2	3450	461.5C	....	....	288	354	204	11	....	80	662c	712c	464	48	....	....	1 7/8	....	4 1/2	5 5/8 × 7/16 × 5/32
		4	1750	461C	287	353	272	338	188	11	4	80	677	727	464	48	15	8×7	1 7/8	5 1/2×10	6 11/16	5 5/8 × 7/16 × 5/32
		6	1160	480C	331	397	316	382	232	11	4	80	759	809	536	48	25	9×8	1 7/8	7×10	8	6 7/8 × 9/16 × 3/16
		8	870	566C	372	438	355	421	271	12	5	110	1005	1055	735	80	40	10×9	2 3/8	7×12	8	6 7/8 × 9/16 × 3/16
		10	690	586C	462	528	443	509	359	12	7	110	1116	1166	836	80	50	11×10	2 3/8	9×12	8	6 7/8 × 9/16 × 3/16
		12	575	642C	501	567	478	544	394	16	7	115	1667	1717	1302	165	50	11×10	2 7/8	9×14	9 1/2	8 1/4 × 11/16 × 1/4
		14	490	644C	561	627	534	600	450	16	11	115	1759	1809	1369	165	75	12×12	2 7/8	9×14	9 1/2	8 1/4 × 11/16 × 1/4
		16	435	750C	611	677	579	645	495	21	11	120	2480	2550	2090	165	75	12×12	3 3/8	12×14	10 1/4	8 7/8 × 13/16 × 9/32
	2200	4	1750	565C	461	861	445	845	295	12	4	110	1135	2335	740	80	15	8×7	1 7/8	5 1/2×10	8	5 5/8 × 7/16 × 5/32
		6	1160	567C	516	916	500	900	350	12	4	110	1168	2368	763	80	25	9×8	1 7/8	7×10	8	6 7/8 × 9/16 × 3/16
		8	870	642C	565	965	544	944	394	16	5	115	1807	3007	1302	165	40	10×9	2 3/8	7×12	9 1/2	6 7/8 × 9/16 × 3/16
25	220 440 550	2	3450	473.5C	....	....	324	388	238	11	....	80	711c	761c	513	48	....	....	1 7/8	....	4 1/2	5 5/8 × 7/16 × 5/32
		4	1750	473C	318	382	303	367	217	11	4	80	718	768	495	48	25	9×8	1 7/8	6×10	6 11/16	5 5/8 × 7/16 × 5/32
		6	1160	567C	369	433	352	416	266	12	5	110	1033	1083	763	80	40	10×9	2 3/8	7×12	8	6 7/8 × 9/16 × 3/16
		8	870	586C	415	479	396	460	310	12	7	110	1116	1166	836	80	50	11×10	2 3/8	8×12	8	6 7/8 × 9/16 × 3/16
		10	690	644C	511	575	488	552	402	16	7	115	1734	1784	1369	165	50	11×10	2 7/8	9×14	9 1/2	8 1/4 × 11/16 × 1/4
		12	575	750C	562	626	530	594	444	21	11	120	2480	2530	2090	165	75	12×12	3 3/8	12×14	10 1/4	8 7/8 × 13/16 × 9/32
		14	490	750C	628	692	593	657	507	21	14	120	2500	2550	2090	165	95	14×12	3 3/8	12×14	10 1/4	8 7/8 × 13/16 × 9/32
		16	435	752C	678	742	643	707	557	21	14	120	2527	2577	2117	165	95	14×12	3 3/8	12×14	10 1/4	8 7/8 × 13/16 × 9/32
	2200	4	1750	565C	501	901	485	885	330	12	4	110	1145	2345	740	80	25	9×8	2 3/8	6×10	8	6 7/8 × 9/16 × 3/16
		6	1160	642C	552	952	531	931	376	16	5	115	1807	3007	1302	165	40	10×9	2 7/8	7×12	9 1/2	8 1/4 × 11/16 × 1/4
		8	870	644C	610	1010	587	987	432	16	7	115	1884	3084	1369	165	50	11×10	2 7/8	9×12	9 1/2	8 1/4 × 11/16 × 1/4
30	220	2	3450	473.5C	....	....	368	430	280	11	....	80	711c	761c	713	48	....	....	1 7/8	....	4 1/2	5 5/8 × 7/16 × 5/32
		4	1750	481C	348	410	333	395	245	11	4	80	751	801	528	48	25	9×8	1 7/8	7×10	6 11/16	5 5/8 × 7/16 × 5/32
		6	1160	567C	404	466	385	447	297	12	7	110	1053	1103	763	80	50	11×10	2 3/8	8×12	8	6 7/8 × 9/16 × 3/16
		8	870	642C	458	520	435	497	347	16	7	115	1667	1717	1302	165	50	11×10	2 7/8	9×12	9 1/2	8 1/4 × 11/16 × 1/4
		10	690	644C	559	621	532	594	444	16	11	115	1759	1809	1369	165	75	12×12	2 7/8	10×14	9 1/2	8 1/4 × 11/16 × 1/4
		12	575	750C	614	676	579	641	491	21	14	120	2500	2550	2090	165	95	14×12	3 3/8	12×14	10 1/4	8 7/8 × 13/16 × 9/32
		14	490	752C	683	745	648	710	560	21	14	120	2527	2577	2117	165	95	14×12	3 3/8	12×14	10 1/4	8 7/8 × 13/16 × 9/32
		16	435	752C	744	806	704	766	616	21	19	120	2555	2605	2115	165	125	16×13	3 3/8	12×14	10 1/4	8 7/8 × 13/16 × 9/32
	440 550	2	3450	473.5C	....	....	368	430	280	11	....	80	711c	761c	513	48	....	....	1 7/8	....	4 1/2	5 5/8 × 7/16 × 5/32
		4	1750	481C	348	410	333	395	245	11	4	80	751	801	528	48	25	9×8	1 7/8	7×10	6 11/16	5 5/8 × 7/16 × 5/32
		6	1160	567C	404	466	385	447	297	12	7	110	1053	1103	763	80	50	11×10	2 3/8	8×12	8	6 7/8 × 9/16 × 3/16
		8	870	642C	458	520	435	497	347	16	7	115	1667	1717	1302	165	50	11×10	2 7/8	9×12	9 1/2	8 1/4 × 11/16 × 1/4
		10	690	644C	559	621	532	594	444	16	11	115	1759	1809	1369	165	75	12×12	2 7/8	10×14	9 1/2	8 1/4 × 11/16 × 1/4
		12	575	750C	614	676	579	641	491	21	14	120	2500	2550	2090	165	95	14×12	3 3/8	12×14	10 1/4	8 7/8 × 13/16 × 9/32
		14	490	752C	683	745	648	710	560	21	14	120	2527	2577	2117	165	95	14×12	3 3/8	12×14	10 1/4	8 7/8 × 13/16 × 9/32
		16	435	752C	744	806	704	766	616	21	19	120	2555	2605	2117	165	125	16×13	3 3/8	12×14	10 1/4	8 7/8 × 13/16 × 9/32
	2200	4	1750	565C	515	910	499	894	339	12	4	110	1145	2345	740	80	25	9×8	2 3/8	6×12	8	6 7/8 × 9/16 × 3/16
		6	1160	642C	595	990	572	967	412	16	7	115	1817	3017	1302	165	50	11×10	2 7/8	8×12	9 1/2	8 1/4 × 11/16 × 1/4
		8	870	644C	650	1045	627	1022	467	16	7	115	1884	3084	1369	165	50	11×10	2 7/8	9×12	9 1/2	8 1/4 × 11/16 × 1/4
		10	690	658C	770	1165	743	1138	583	16	11	115	2156	3456	1716	165	75	12×12	2 7/8	11×14	9 1/2	8 1/4 × 11/16 × 1/4

For notes and explanation of reference letters, see last page of this table.

TABLE 152—*Continued*

Horse-power	Volts	Number of Poles	Full Load, R.p.m.	Frame	Price, Dollars: Complete motor with Type "A" manual starter Class 10,700	Price, Dollars: Complete motor with Magnetic auto-transformer type starter	Price, Dollars: Bare motor with Type "A" manual starter Class 10,700	Price, Dollars: Bare motor with Magnetic auto-transformer type starter	Price, Dollars: Bare motor	Price, Dollars: Omission or addition for Rails	Price, Dollars: Omission or addition for Pulley	Price, Dollars: Addition for vertical parts	Approximate Shipping Weight, Pounds: Complete motor with Type A starter	Approximate Shipping Weight, Pounds: Complete motor with Magnetic starter	Approximate Shipping Weight, Pounds: Bare motor	Approximate Shipping Weight, Pounds: Omission or addition for Rails	Approximate Shipping Weight, Pounds: Omission or addition for Pulley	Pulley Dimensions, Inches: Standard diameter and face	Pulley Dimensions, Inches: Bore	Pulley Dimensions, Inches: Special pulley, minimum diameter and maximum face	Shaft Dimensions, Inches: Standard shaft extension	Shaft Dimensions, Inches: Standard keyway dimensions, L. W. D.
40	220	2	3450	583.5C	.....	.....	455	515	330	12	.....	110	1984c	1159c	804	80	.....	.....	$2\frac{3}{8}$	.....	5	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		4	1750	583C	440	500	423	483	298	12	5	110	1124	1199	804	80	40	10×9	$2\frac{3}{8}$	$7\frac{1}{2}\times12$	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		6	1160	587C	498	558	479	539	354	12	7	110	1237	1312	907	80	50	11×10	$2\frac{3}{8}$	10×12	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		8	870	644C	568	628	541	601	416	16	11	115	1809	1864	1369	165	75	12×12	$2\frac{7}{8}$	10×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		10	690	750C	680	740	645	705	520	21	14	120	2550	2625	2090	165	95	14×12	$3\frac{3}{8}$	11×14	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		12	575	752C	734	794	699	759	574	21	14	120	2577	2652	2117	165	95	14×12	$3\frac{3}{8}$	12×14	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		14	490	754C	820	880	780	840	655	21	19	120	2748	2823	2158	165	125	16×13	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		16	435	774C	891	951	845	905	720	21	25	120	3131	3206	2596	165	170	18×15	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
	440 550	2	3450	583.5C	.....	.....	420	492	330	12	.....	110	1084c	1159c	804	80	.....	.....	$2\frac{3}{8}$	.....	5	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		4	1750	583C	405	477	388	460	298	12	5	110	1124	1199	804	80	40	10×9	$2\frac{3}{8}$	$7\frac{1}{2}\times12$	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		6	1160	587C	463	535	444	516	354	12	7	110	1237	1312	907	80	50	11×10	$2\frac{3}{8}$	10×12	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		8	870	644C	533	605	506	578	416	16	11	115	1809	1864	1369	165	75	12×12	$2\frac{7}{8}$	10×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		10	690	750C	645	717	610	682	520	21	14	120	2550	2625	2090	165	95	14×12	$3\frac{3}{8}$	11×14	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		12	575	752C	699	771	664	736	574	21	14	120	2570	2652	2117	165	95	14×12	$3\frac{3}{8}$	12×14	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		14	490	754C	785	857	745	817	655	21	19	120	2748	2823	2158	165	125	14×12	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		16	435	774C	856	928	810	882	720	21	25	120	3131	3206	2596	165	170	18×15	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
	2200	4	1750	583C	571	981	554	934	389	12	5	110	1224	2424	804	80	40	10×9	$2\frac{3}{8}$	$7\frac{1}{2}\times12$	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		6	1160	642C	662	1072	639	1049	474	16	7	115	1817	3017	1302	165	50	11×10	$2\frac{7}{8}$	10×12	$9\frac{1}{2}$	$6\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		8	870	652C	724	1134	697	1107	532	16	11	115	2080	3280	1540	165	75	12×12	$2\frac{7}{8}$	10×12	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		10	690	750C	845	1255	810	1220	645	21	14	120	2650	3850	2090	165	95	14×12	$3\frac{3}{8}$	11×12	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
	220	2	3450	587.5C	.....	.....	520	590	390	12	.....	110	1187c	1262c	907	80	.....	.....	$2\frac{3}{8}$	.....	5	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		4	1750	587C	496	566	477	547	347	12	7	110	1237	1312	907	80	50	11×10	$2\frac{3}{8}$	.....	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		6	1160	642C	567	637	540	610	410	16	11	115	1742	1817	1302	165	75	12×12	$2\frac{7}{8}$	19×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		8	870	644C	639	709	609	679	479	16	14	115	1829	1904	1369	165	95	14×12	$2\frac{7}{8}$	11×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		10	690	752C	757	827	717	787	587	21	19	120	2607	2682	2117	165	125	16×13	$3\frac{3}{8}$	12×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		12	575	752C	820	890	780	850	650	21	19	120	2607	2682	2117	165	125	16×13	$3\frac{3}{8}$	12×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		14	490	762C	918	988	872	942	742	21	25	120	2849	2924	2314	165	170	18×15	$3\frac{3}{8}$	15×14	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		16	435	774C	991	1061	945	1015	815	21	25	120	3031	3406	2596	165	170	18×15	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$

50	440, 550	2	3450	587.5C	......	......	484	555	390	12	......	110	1187c	1262c	907	80	......	......	$2\frac{3}{8}$	......	5	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		4	1750	587C	460	531	441	512	347	12	7	110	1237	1312	907	80	50	11×10	$2\frac{3}{8}$	......	8	$6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$
		6	1160	642C	531	602	504	575	410	16	11	115	1742	1817	1302	165	75	12×12	$2\frac{7}{8}$	10×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		8	870	644C	603	674	573	644	479	16	14	115	1829	1904	1369	165	95	14×12	$2\frac{7}{8}$	11×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		10	690	752C	721	792	681	752	587	21	19	120	2607	2682	2117	165	125	16×13	$3\frac{3}{8}$	12×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		12	575	752C	784	855	744	815	650	21	19	120	2607	2682	2117	165	125	16×13	$3\frac{3}{8}$	12×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		14	490	762C	882	953	836	907	742	21	25	120	2849	2924	2314	165	170	18×15	$3\frac{3}{8}$	15×14	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		16	435	774C	955	1026	909	980	815	21	25	120	3131	3406	2596	165	170	18×15	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
	2200	4	1750	643C	629	1034	606	1011	436	16	7	115	1929	3129	1369	165	50	11×10	$2\frac{7}{8}$	......	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		6	1160	646C	729	1134	702	1107	532	16	11	125	2058	3258	1518	165	75	12×12	$2\frac{7}{8}$	11×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		8	870	752C	793	1198	758	1163	588	21	14	120	2577	3877	2117	165	95	14×12	$3\frac{3}{8}$	12×14	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		10	690	752C	912	1317	872	1277	702	21	19	120	2807	4007	2117	165	125	16×13	$3\frac{3}{8}$	14×15	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		12	575	774C	964	1369	924	1329	754	21	19	110	3086	4286	2596	165	125	16×13	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
60	220	4	1750	645C	......	......	521	843	393	16	......	115	2155c	2455c	1590	165	......	......	$2\frac{7}{8}$	......	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		6	1160	644C	616	938	589	911	461	16	11	115	2009	2309	1369	165	75	12×12	$2\frac{7}{8}$	11×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		8	870	662C	697	1019	667	989	539	16	14	115	2377	2677	1717	165	95	14×12	$2\frac{7}{8}$	12×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		10	690	752C	818	1140	778	1100	650	21	19	120	2817	3117	2127	165	125	16×13	$3\frac{3}{8}$	14×15	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		12	575	774C	895	1217	849	1171	721	21	25	120	3331	3631	2596	165	170	18×15	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		14	490	856C	1022	1344	953	1275	825	44	25	230	4244	3544	3344	330	170	18×15	$3\frac{7}{8}$	16×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		16	435	856C	1109	1431	1038	1360	910	44	27	230	4244	3544	3344	330	170	20×15	$3\frac{7}{8}$	16×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
	440, 550	4	1750	645C	......	......	521	649	393	16	......	115	2105c	2155c	1590	165	......	......	$2\frac{7}{8}$	......	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		6	1160	644C	616	744	589	717	461	16	11	115	1959	2009	1369	165	75	12×12	$2\frac{7}{8}$	11×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		8	870	662C	697	825	667	795	539	16	14	115	2327	2377	1717	165	95	14×12	$2\frac{7}{8}$	12×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		10	690	752C	818	946	778	906	650	21	19	120	2767	2817	2127	165	125	16×13	$3\frac{3}{8}$	14×15	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		12	575	774C	895	1023	849	977	721	21	25	120	3281	3331	2596	165	170	18×15	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		14	490	856C	1022	1150	953	1081	825	44	25	230	4194	4244	3344	330	170	18×15	$3\frac{7}{8}$	16×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		16	435	856C	1109	1237	1038	1166	910	44	27	230	4194	4244	3344	330	170	20×15	$3\frac{7}{8}$	16×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
	2200	4	1750	653C	......	......	648	1029	482	16	......	115	2152c	3277c	1562	165	......	......	$2\frac{7}{8}$	......	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		6	1160	664C	777	1158	750	1131	584	16	11	115	2425	3558	1768	165	75	12×12	$2\frac{7}{8}$	11×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		8	870	762C	846	1227	811	1192	645	21	14	120	2999	4124	2314	165	95	14×12	$3\frac{3}{8}$	12×14	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		10	690	762C	960	1341	920	1301	754	21	19	120	3029	4154	2314	165	125	16×13	$3\frac{3}{8}$	12×15	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		12	575	854C	1046	1427	977	1358	811	44	25	120	4203	5328	3278	330	170	18×15	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
75	220	4	1750	645C	......	......	594	936	461	16	......	115	2055c	2355c	1490	165	......	......	$2\frac{7}{8}$	......	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		6	1160	662C	700	1042	670	1012	537	16	14	115	2177	2677	1717	165	95	14×12	$2\frac{7}{8}$	13×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		8	870	752C	799	1141	759	1101	626	21	19	120	2807	3107	2117	165	125	16×13	$3\frac{3}{8}$	14×16	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		10	690	754C	913	1255	867	1209	734	21	25	120	2893	3193	2158	165	170	18×15	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		12	575	774C	995	1337	949	1291	816	21	25	120	3331	3631	2596	165	170	18×15	$3\frac{3}{8}$	16×20	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		14	490	854C	1140	1482	1069	1411	936	44	27	230	4244	4544	3344	330	170	20×15	$3\frac{7}{8}$	16×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		16	435	874C	1241	1583	1168	1510	1035	44	29	230	4546	4846	3586	330	230	21×17	$3\frac{7}{8}$	17×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
	440, 550	4	1750	645C	......	......	594	717	461	16	......	115	2005c	2055c	1490	165	......	......	$2\frac{7}{8}$	......	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		6	1160	662C	700	823	670	793	537	16	14	115	2327	2377	1717	165	95	14×12	$2\frac{7}{8}$	13×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		8	870	752C	799	922	759	882	626	21	19	120	2757	2807	2117	165	125	16×13	$3\frac{3}{8}$	14×16	$10\frac{1}{4}$	$8\frac{1}{8}\times\frac{13}{16}\times\frac{9}{32}$
		10	690	754C	913	1036	867	990	734	21	25	120	2843	2893	2158	165	170	18×15	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		12	575	774C	995	1118	949	1072	816	21	25	120	3281	3331	2596	165	170	18×15	$3\frac{3}{8}$	16×20	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		14	490	854C	1136	1259	1065	1188	936	44	27	230	4194	4244	3344	330	170	20×15	$3\frac{7}{8}$	16×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		16	435	874C	1231	1354	1158	1281	1035	44	29	230	4496	4546	3586	330	230	21×17	$3\frac{7}{8}$	17×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$

For notes and explanation of reference letters, see last page of this table.

TABLE 152—*Continued*

Horse-power	Volts	Number of Poles	Full Load, R.p.m.	Frame	Price, Dollars								Approximate Shipping Weight, Pounds					Pulley Dimensions, Inches			Shaft Dimensions, Inches	
					Complete motor with		Bare motor with			Omission or addition for			Complete motor with			Omission or addition for						
					Type "A" manual starter Class 10,700	Magnetic autotransformer type starter	Type "A" manual starter Class 10,700	Magnetic autotransformer type starter	Bare motor	Rails	Pulley	Addition for vertical parts	Type A starter	Magnetic starter	Bare motor	Rails	Pulley	Standard diameter and face	Bore	Special pulley, minimum diameter and maximum face	Standard shaft extension	Standard keyway dimensions, L. W. D.
75	2200	4	1750	653C	......	......	718	1098	547	16	......	115	2252*c*	3377*c*	1562	165	......	......	$2\frac{7}{8}$	......	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		6	1160	664C	860	1240	830	1210	659	16	14	115	2453	3578	1768	165	95	14×12	$2\frac{7}{8}$	13×14	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		8	870	762C	934	1314	894	1274	723	21	19	120	3029	4154	2314	165	125	16×13	$3\frac{3}{8}$	15×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		10	690	774C	1043	1423	997	1377	826	21	25	120	3356	4481	2596	165	170	18×15	$3\frac{3}{8}$	16×20	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{15}{16}\times\frac{9}{32}$
		12	575	854C	1135	1515	1066	1446	895	44	25	230	4203	4328	3278	330	170	18×15	$3\frac{7}{8}$	16×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
100	220	4	1750	663C	......	......	764	1054	574	16	......	115	2334*c*	2634*c*	1769	165	......	......	$2\frac{7}{8}$	......	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		6	1160	752C	902	1192	862	1152	672	21	19	120	2807	3107	2117	165	125	16×13	$3\frac{3}{8}$	13×14	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		8	870	754C	997	1287	951	1241	761	21	25	120	2893	3193	2158	165	170	18×15	$3\frac{3}{8}$	14×16	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		10	590	854C	1129	1419	1058	1348	868	44	27	230	4178	4478	3278	330	170	20×15	$3\frac{7}{8}$	15×18	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		12	575	854C	1226	1516	1153	1443	963	44	29	230	4238	4538	3278	330	230	21×17	$3\frac{7}{8}$	16×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		14	490	876C	1373	1663	1290	1580	1100	44	39	230	4910	5210	3740	330	440	25×17	$3\frac{7}{8}$	22×22	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		16	435	936	1492	1782	1410	1700	1220	53	29	......	5880	5580	4070	880	230	21×17	$3\frac{7}{8}$	20×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
	440 550	4	1750	663C	......	......	712	830	574	16	......	115	2284*c*	2334*c*	1769	165	......	......	$2\frac{7}{8}$	......	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		6	1160	752C	850	968	810	928	672	21	19	120	2757	2807	2117	165	125	16×13	$3\frac{3}{8}$	13×14	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		8	870	754C	945	1063	899	1017	761	21	25	120	2843	2893	2158	165	170	18×15	$3\frac{3}{8}$	14×16	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		10	690	854C	1077	1195	1006	1124	868	44	27	230	4128	4178	3278	330	170	20×15	$3\frac{7}{8}$	15×18	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		12	575	854C	1174	1292	1101	1219	963	44	29	230	4188	4238	3278	330	230	21×17	$3\frac{7}{8}$	16×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		14	490	876C	1321	1439	1238	1356	1100	44	39	230	4860	4910	3740	330	440	25×17	$3\frac{7}{8}$	22×22	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		16	435	936	1430	1558	1358	1476	1220	53	29	......	5880	5580	4070	880	230	21×17	$3\frac{7}{8}$	20×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
	2200	4	1750	663C	......	......	829	1213	653	16	......	115	1959*c*	3484*c*	1769	165	......	......	$2\frac{7}{8}$	......	$9\frac{1}{2}$	$8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$
		6	1160	760C	993	1377	953	1337	777	21	19	120	2915	4040	2200	165	125	16×13	$3\frac{3}{8}$	13×16	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		8	870	774C	1069	1453	1023	1407	847	21	25	120	3356	4481	2596	165	170	18×15	$3\frac{3}{8}$	16×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		10	690	856C	1183	1567	1112	1496	936	44	27	230	4269	5394	3344	330	170	20×15	$3\frac{7}{8}$	17×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		12	575	874C	1273	1657	1200	1584	1024	44	29	230	4571	5696	3586	330	230	21×17	$3\frac{7}{8}$	19×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		14	490	938	1398	1782	1316	1700	1140	53	29	......	6928	5803	4268	880	230	21×17	$3\frac{7}{8}$	20×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{3}{16}$
		16	435	938	1518	1902	1436	1820	1260	53	29	......	6928	5803	4268	880	230	21×17	$3\frac{7}{8}$	20×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$

125	220	4	1750	761C	......	......	880	1388	680	21	......	120	3883c	3983c	2468	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		6	1760	764C	......	......	990	1498	790	21	......	120	3915c	4015c	2500	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		8	870	774C	1141	1649	1093	1601	893	21	27	120	4181	4281	2596	165	170	20×15	$3\frac{3}{8}$	18×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		10	690	856C	1269	1777	1186	1694	986	44	39	230	5364	5464	3344	330	440	25×17	$3\frac{7}{8}$	20×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		12	575	874C	1381	1889	1298	1806	1098	44	39	230	5606	5706	3586	330	440	25×17	$3\frac{7}{8}$	20×20	$13\frac{3}{4}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		14	490	936	1510	2018	1420	1928	1220	53	37	......	6430	6530	4970	880	230	23×19	$3\frac{7}{8}$	22×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
		16	435	938	1634	2142	1540	2048	1340	53	41	......	6838	6938	4268	880	440	27×17	$3\frac{7}{8}$	24×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
	440 550	4	1750	761C	......	......	823	1170	680	21	......	120	3033c	3333c	2468	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		6	1160	764C	......	......	933	1280	790	21	......	120	3065c	3365c	2500	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		8	870	774C	1084	1431	1036	1383	893	21	27	120	3331	3631	2596	165	170	20×15	$3\frac{3}{8}$	18×18	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		10	690	856C	1212	1559	1129	1476	986	44	39	230	4514	4814	3344	330	440	25×17	$3\frac{7}{8}$	20×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		12	575	874C	1324	1617	1241	1588	1098	44	39	230	4756	5056	3586	330	440	25×17	$3\frac{7}{8}$	20×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		14	490	936	1453	1800	1363	1710	1220	53	37	......	5580	5880	4070	880	230	23×19	$3\frac{7}{8}$	22×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
		16	435	938	1577	1924	1483	1830	1340	53	41	......	5988	6288	4268	880	440	27×17	$3\frac{7}{8}$	24×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
	2200	4	1750	761C	......	......	942	1327	752	21	......	120	3108c	4233c	2468	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		6	1160	772C	......	......	1080	1465	890	21	......	120	3072c	4197c	2452	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		8	870	856C	1228	1613	1157	1542	967	44	27	230	3319	4444	3344	330	170	20×15	$3\frac{7}{8}$	16×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		10	690	876C	1306	1691	1233	1618	1043	44	29	230	4775	5930	3740	330	230	21×17	$3\frac{7}{8}$	20×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		12	575	938	1429	1814	1337	1722	1147	53	39	275	5063	7188	4268	880	440	25×17	$3\frac{7}{8}$	20×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
		14	490	938	1550	1935	1460	1845	1270	53	37	......	5853	6978	4268	880	230	23×19	$3\frac{7}{8}$	22×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
		16	435	1000	1684	2069	1580	1965	1390	63	41	......	6795	7920	5830	1050	440	27×17	$3\frac{7}{8}$	24×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
150	220	4	1750	771.1C	......	......	1163	1586	783	21	......	120	3850c	4000c	2435	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		6	1160	772C	......	......	1298	1721	918	21	......	120	3847c	3997c	2432	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		8	870	854C	1478	1901	1405	1828	1025	44	29	230	5088	5238	3278	330	230	21×17	$3\frac{7}{8}$	19×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		10	690	874C	1561	1984	1478	1901	1098	44	39	230	5606	5756	3586	330	440	25×17	$3\frac{7}{8}$	22×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		12	575	876C	1685	2108	1602	2025	1222	44	39	230	5760	5910	3740	330	440	25×17	$3\frac{7}{8}$	22×22	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		14	490	938	1813	2236	1720	2143	1340	53	40	......	6838	6988	4268	880	440	28×17	$3\frac{7}{8}$	25×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
		16	435	1000	1943	2366	1840	2263	1460	63	40	......	8570	8720	5830	1050	440	28×20	$4\frac{7}{8}$	25×24	16	$15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
	440 550	4	1750	771.1C	......	......	930	1283	783	21	......	120	3300c	3300c	2435	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		6	1160	772C	......	......	1065	1418	918	21	......	120	2997c	3297c	2432	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		8	870	854C	1245	1598	1172	1525	1025	44	29	230	4238	4538	3278	330	230	21×17	$3\frac{7}{8}$	19×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		10	690	874C	1328	1681	1245	1598	1098	44	39	230	4756	5056	3586	330	440	25×17	$3\frac{7}{8}$	22×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		12	575	876C	1452	1805	1369	1722	1222	44	39	230	4910	5210	3740	330	440	25×17	$3\frac{7}{8}$	22×22	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		14	490	938	1580	1933	1487	1840	1340	53	40	......	4988	6288	4268	880	440	28×17	$3\frac{7}{8}$	25×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
		16	435	1000	1710	2063	1607	1960	1460	63	40	......	7720	8020	5830	1050	440	28×20	$4\frac{7}{8}$	25×24	16	$15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
	2200	4	1750	771.1C	......	......	1052	1427	852	21	......	120	3075c	4300c	2435	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$
		6	1160	772C	......	......	1204	1579	1004	21	......	120	3072	4297c	2432	165	......	......	$3\frac{3}{8}$	......	$10\frac{1}{4}$	$8\frac{7}{8}\times\frac{15}{16}\times\frac{9}{32}$
		8	870	870C	1358	1733	1285	1660	1085	44	29	230	4497	5722	3462	330	230	21×17	$3\frac{7}{8}$	19×20	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		10	690	876C	1430	1805	1347	1722	1147	44	39	275	4985	6210	3940	330	440	25×17	$3\frac{7}{8}$	22×20	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
		12	575	954	1554	1929	1467	1842	1267	53	34	275	6388	7613	4593	880	440	25×17	$3\frac{7}{8}$	22×22	$13\frac{1}{2}$	$11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$
		14	490	954A	1678	2053	1590	1965	1390	53	35	......	6388	7613	4593	880	440	27×19	$4\frac{3}{8}$	22×23	$15\frac{1}{4}$	$13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
		16	435	1000	1813	2188	1710	2085	1510	63	40	......	7795	9020	5830	1050	440	28×20	$4\frac{7}{8}$	25×24	16	$15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$

For notes and explanation of reference letters, see last page of this table.

TABLE 152—*Concluded*

Horse-power	Volts	Number of Poles	Full Load, R.p.m.	Frame	Price, Dollars: Complete motor with Type "A" manual starter Class 10,700	Price, Dollars: Complete motor with Magnetic auto-transformer type starter	Price, Dollars: Bare motor with Type "A" manual starter Class 10,700	Price, Dollars: Bare motor with Magnetic auto-transformer type starter	Price, Dollars: Bare motor	Price, Dollars: Omission or addition for Rails	Price, Dollars: Omission or addition for Pulley	Price, Dollars: Addition for vertical parts	Approximate Shipping Weight, Pounds: Complete motor with Type A starter	Approximate Shipping Weight, Pounds: Complete motor with Magnetic starter	Approximate Shipping Weight, Pounds: Bare motor	Approximate Shipping Weight, Pounds: Omission or addition for Rails	Approximate Shipping Weight, Pounds: Omission or addition for Pulley	Pulley Dimensions, Inches: Standard diameter and face	Pulley Dimensions, Inches: Bore	Pulley Dimensions, Inches: Special pulley, minimum diameter and maximum face	Shaft Dimensions, Inches: Standard shaft extension	Shaft Dimensions, Inches: Standard keyway dimensions, L. W. D.
200	220	4	1750	855C	......	......	1755	2000	985	44	......	230	5005c	5005c	3225	330	......	......	$3\frac{7}{8}$	......	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		6	1160	871C	......	......	1952	2197	1182	44	......	230	5300c	5300c	3520	330	......	......	$3\frac{7}{8}$	......	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		8	870	874C	......	......	2044	2289	1274	44	......	230	5366c	5366c	3586	330	......	......	$3\frac{7}{8}$	......	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		10	690	954A	2161	2406	2070	2315	1300	53	38	275	7363	7363	4593	880	440	28×18	$4\frac{3}{8}$	23×22	$15\frac{1}{4}$	$13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
		12	575	954A	2312	2557	2219	2464	1449	53	40	275	7363	7363	4593	880	440	32×20	$4\frac{3}{8}$	25×24	$15\frac{1}{4}$	$13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
		14	490	1000	2458	2703	2345	2590	1575	63	50	......	8770	8770	5830	1050	440	33×20	$4\frac{7}{8}$	28×24	16	$15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
		16	435	1000	2593	2838	2470	2715	1700	63	60	......	8960	8960	5830	1050	630	38×20	$4\frac{7}{8}$	32×24	16	$15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
	440 550	4	1750	871C	......	......	1171	1494	985	44	......	230	3955c	4255c	3225	330	......	......	$3\frac{7}{8}$	......	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		6	1160	873C	......	......	1368	1691	1182	44	......	230	4250c	4550c	3520	330	......	......	$3\frac{7}{8}$	......	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		8	870	874C	......	......	1460	1783	1274	44	......	230	4316c	4616c	3586	330	......	......	$3\frac{7}{8}$	......	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		10	690	954A	1577	1900	1486	1809	1300	53	38	275	6313	6613	4593	880	440	28×18	$4\frac{3}{8}$	23×22	$15\frac{1}{4}$	$13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
		12	575	954A	1728	2051	1635	1958	1449	53	40	275	6313	6613	4593	880	440	32×20	$4\frac{3}{8}$	25×24	$15\frac{1}{4}$	$13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
		14	490	1000	1874	2197	1761	2084	1575	63	50	......	7720	8020	5830	1050	440	33×20	$4\frac{7}{8}$	28×24	16	$15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
		16	435	1002	2009	2332	1886	2209	1700	63	60	......	7910	8210	5830	1050	630	38×20	$4\frac{7}{8}$	32×24	16	$15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
	2200	4	1750	871C	......	......	1242	1627	1033	44	......	230	4391c	5616c	3586	330	......	......	$3\frac{7}{8}$	......	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		6	1160	873C	......	......	1440	1825	1231	44	......	230	4435c	5660c	3630	330	......	......	$3\frac{7}{8}$	......	$13\frac{1}{2}$	$12\times\frac{15}{16}\times\frac{5}{16}$
		8	870	939A	1608	1993	1522	1907	1313	53	33	275	6305	7530	4510	880	440	22×20	$4\frac{3}{8}$	20×22	$15\frac{1}{4}$	$13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
		10	690	956A	1646	2031	1548	1933	1339	53	45	275	6470	7695	4672	880	440	28×18	$4\frac{3}{8}$	23×22	$15\frac{1}{4}$	$13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
		12	575	1002	1817	2202	1695	2080	1486	63	59	460	8095	9320	5940	1050	630	32×20	$4\frac{7}{8}$	25×24	16	$15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
		14	490	1000	1947	2332	1819	2204	1610	63	65	......	7985	9210	5830	1050	630	33×20	$4\frac{7}{8}$	28×24	16	$15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$
		16	435	1002	2072	2457	1939	2324	1730	63	70	......	8165	9390	5940	1050	700	38×20	$4\frac{7}{8}$	32×24	16	$15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$

NOTES AND EXPLANATION OF REFERENCE LETTERS

a Class 200 and 300 frames are supplied with bedplate instead of rails.

b Pulleys require special offset hub in most cases, if wider pulley is used, special shaft extension is required.

c Pulleys are not included in weight.

d Length of shaft extension, pulley end.

e Not recommended for belted service.

f Price includes a class 10,700, type A enclosed auto-starter providing overload and low voltage protection by automatically resetting relays.

g (220, 440 and 550 volts). Price includes a class 11,500, type AF enclosed magnetic auto-starter, "start" and "stop" push button station and "reset" push button. This starter provides overload protection and low-voltage release or protection. When ordering push button station specify whether low-voltage protection or low-voltage release is desired. The "start" and "stop" push button station may be omitted at $5.00 if another type of master switch is desired.

h (2200 volts). Price includes a Class 11,800 magnetic starter and "start" and "stop" push button station. This starter provides overload protection and low-voltage release or protection. When ordering push button station specify whether low-voltage protection or low-voltage release is desired. "Start" and "stop" push button station may be omitted at $2.50 if another type of master switch is desired.

i (220 volts). Price includes a Class 11,550 enclosed magnetic starter and "start" and "stop" push button station. This starter provides overload protection and low-voltage release or protection. When ordering push button specify whether low-voltage protection or low-voltage release is desired. The "start" and "stop" push button station may be omitted at $2.50 if another type of master switch is desired.

j (440 and 550 volts). Price includes a Class 11,500 AF enclosed magnetic auto-starter, "start" and "stop" push button station and "reset" push button. This starter provides overload protection and low-voltage release or protection. When ordering push buttons specify whether low-voltage protection or low-voltage release is desired. The "start" and "stop" push button station may be omitted at $5.00 if another type of master switch is desired.

k (2200 volts). Price includes a Class 11,800 magnetic starter and "start" and "stop" push button station. This starter provides overload protection. When ordering push button station specify whether low-voltage protection or low-voltage release is desired. "Start" and "stop" push button may be omitted at $2.50 if another type of master switch is desired.

l (220 volts). Price includes a Class 10,800 auto-transformer type starting panel with oil circuit-breakers for starting and running and separately mounted auto-transformer. This starter can be supplied for 3-phase circuits only, and provides overload and low voltage protection. Ammeter may be added at $43.00.

m (440, 550 and 2200 volts). Price includes a Class 10,700 type A enclosed auto-starter providing overload and low-voltage protection by automatically resetting relays.

n (220 volts). Price includes a Class 11,750 magnetic starter and "start" and "stop" push button station. This starter provides overload protection and low-voltage release or protection. When ordering push button specify whether low-voltage protection or low-voltage release is desired. The "start" and "stop" push button station may be omitted at $2.50 if another type of master switch is desired

o (440 and 550). Price includes a Class 11,550 enclosed magnetic starter and "start" and "stop" push button station. This starter provides overload protection and low-voltage release or protection. When ordering push button specify whether low-voltage protection or low-voltage release is desired. The "start" and "stop" push button may be omitted at $2.50 if another type of master switch is desired

p Special offset hub pulley.

Conduit boxes are regularly furnished for motor frames
- 200C to 600C inclusive.
- 400C to 700C inclusive.
- 600C to 800C inclusive.
- 700C to 800C inclusive. No allowance for omission.
- 900 if requested. No allowance for omission.
- 900 to 1000 if requested.

NOTE—Ball Bearings can be furnished on all motor frames in this table at an addition of 10 per cent to the bare motor price.

TABLE 153

Brine and Water Pumps

(Centrifugal Type)

Model Pump	Horse Power Motor	Pumps: Price of bronze fitted or all iron pump with base and coupling	Pumps: Price of all bronze pump with base and coupling	Motors: Price of induction 2 or 3 phase motors, 60 cycles, 110, 220 or 440 v., 1800 r.p.m.	Motors: Price of repulsion starter induction single phase motors, 60 cycles, 110-220 volts, 1800 r.p.m.	Motors: Price of d.c. constant speed motors, 115-230 volts, 1800 r.p.m.	Starters: Price of starters for a.c. 3 phase motors (for 2 phase, 4 wire add 40 per cent)	Starters: Price of starters for a.c. single phase motors	Starters: Price of starters for d.c. motors	Shipping Weight of Pump, Base and Polyphase Motor in.	Size of Pipe: Size of Suction in.	Size of Pipe: Size of discharge in.
CC	1/6	$75.00	$75.00	....	$31.20	....	....	$30.00	....	215	1	3/4
	1/4	75.00	75.00	....	37.80	$52.00	....	30.00	$49.00	215	1	3/4
	1/2	75.00	75.00	$49.30	53.60	70.40	$30.00	30.00	49.00	215	1	3/4
AAB	1/2	90.00	90.00	49.30	53.60	70.40	30.00	30.00	49.00	225	1	3/4
	3/4	90.00	90.00	58.00	72.50	84.80	30.00	30.00	49.00	235	1	3/4
LP	3/4	105.00	120.00	58.00	72.50	84.80	30.00	30.00	49.00	255	1 1/2	1 1/4
	1	105.00	120.00	63.80	89.20	108.00	30.00	30.00	49.00	260	1 1/2	1 1/4
M-1	3/4	115.00	130.00	58.00	72.50	84.80	30.00	30.00	49.00	240	1	1
	1	115.00	130.00	63.80	89.20	108.00	30.00	30.00	49.00	270	1	1
	1 1/2	115.00	130.00	68.90	110.90	117.60	30.00	30.00	49.00	285	1	1
L	3/4	120.00	135.00	58.00	72.50	84.80	30.00	30.00	49.00	255	1 1/4	1
	1	120.00	135.00	63.80	89.20	108.00	30.00	30.00	49.00	260	1 1/4	1
	1 1/2	120.00	135.00	68.90	110.90	117.60	30.00	30.00	49.00	290	1 1/4	1
MP	1 1/2	130.00	145.00	68.90	110.90	117.60	30.00	30.00	49.00	295	1 1/2	1 1/4
	2	130.00	145.00	76.10	128.30	144.00	30.00	30.00	49.00	310	1 1/2	1 1/4
	3	130.00	145.00	86.30	158.80	169.60	30.00	40.00	49.00	340	1 1/2	1 1/4
N	1 1/2	135.00	150.00	68.90	110.90	117.60	30.00	30.00	53.00	325	1 1/2	1 1/4
	2	135.00	150.00	76.10	128.30	144.00	30.00	30.00	49.00	340	1 1/2	1 1/4
	3	135.00	150.00	86.30	158.80	169.60	30.00	40.00	53.00	370	1 1/2	1 1/4
O	2	140.00	160.00	76.10	128.30	144.00	30.00	30.00	49.00	345	2	1 1/2
	3	140.00	160.00	86.30	158.80	169.60	30.00	40.00	53.00	375	2	1 1/2
	5	140.00	160.00	105.80	208.00	252.80	30.00	40.00	53.00	400	2	1 1/2

P	2	145.00	165.00	76.10	128.30	144.00	30.00	30.00	49.00	350	2	1½
	3	145.00	165.00	86.30	158.80	169.60	30.00	40.00	53.00	380	2	1½
	5	145.00	165.00	105.80	208.00	252.80	30.00	40.00	53.00	405	2	1½
S	3	150.00	175.00	86.30	158.80	169.60	30.00	40.00	53.00	380	2½	2
	5	150.00	175.00	105.80	208.00	252.80	30.00	40.00	53.00	405	2½	2
	7½	150.00	175.00	130.50	284.00	307.20	40.00	50.00	64.00	465	2½	2
LN	3	155.00	180.00	86.30	158.80	169.60	30.00	40.00	53.00	390	2	1½
	5	155.00	180.00	105.80	208.00	252.80	30.00	40.00	53.00	415	2	1½
	7½	155.00	180.00	130.50	284.00	307.20	40.00	50.00	64.00	485	2	1½
T	3	170.00	195.00	86.30	158.80	169.60	30.00	40.00	53.00	390	2½	2
	5	170.00	195.00	105.80	208.00	252.80	30.00	40.00	53.00	415	2½	2
	7½	170.00	195.00	130.50	284.00	307.20	40.00	50.00	64.00	485	2½	2
L-5	3	175.00	205.00	86.30	158.80	169.60	30.00	40.00	53.00	405	3	2½
	5	175.00	205.00	105.80	208.00	252.80	30.00	40.00	53.00	430	3	2½
	7½	175.00	205.00	130.50	284.00	307.20	40.00	50.00	64.00	500	3	2½
LS	7½	275.00	315.00	130.50	284.00	307.20	40.00	50.00	64.00	825	2½	2½
	10	275.00	315.00	155.00	....	352.00	40.00	....	64.00	935	2½	2½
	15	275.00	315.00	198.50	....	456.00	50.00	....	....	975	2½	2½
	20	275.00	315.00	243.50	....	544.00	60.00	....	....	1100	2½	2½
LSS	7½	285.00	325.00	130.50	....	307.20	40.00	....	64.00	885	2½	2½
	10	285.00	325.00	155.00	....	352.00	40.00	....	64.00	1000	2½	2½
	15	285.00	325.00	198.50	....	456.00	50.00	....	....	1050	2½	2½
	20	285.00	325.00	243.50	....	544.00	60.00	....	....	1160	2½	2½
	25	285.00	325.00	290.00	....	595.00	60.00	....	....	1300	2½	2½
M-6	7½	335.00	380.00	130.50	....	307.20	40.00	....	64.00	825	4	3½
	10	335.00	380.00	155.00	....	352.00	40.00	....	64.00	935	4	3½
	15	335.00	380.00	198.50	....	456.00	50.00	....	....	975	4	3½
	20	335.00	380.00	243.50	....	544.00	60.00	....	....	1100	4	3½
H-6	20	550.00	620.00	243.50	....	544.00	60.00	....	....	1250	3½	3
	25	550.00	620.00	290.00	....	595.00	60.00	....	....	1400	3½	3
	30	550.00	620.00	335.00	....	646.00	60.00	....	....	1400	3½	3

All prices are f.o.b. Iowa.

PUMPS.—Bronze fitted have Cast Iron casings with bronze impellers and packing glands with monel metal shafts.

All bronze pumps are of bronze construction throughout with monel metal shafts.

All iron pumps are of cast iron construction throughout with monel metal shafts.

All pumps have two outboard radial thrust ball bearings.

STARTERS.—Polyphase and single phase a.c. motor starters furnished are type enclosed in steel case for wall mounting with conduit knockouts; have thermal inverse time limit overload relays which are hand reset from outside of cabinet and low voltage protection. Prices include one "start" and "stop" push bottom station. Add 40 per cent to price for 2 phase 4 wire. When power company will not permit motor to be thrown directly across the line, then resistance type starters must be used, prices of which will be furnished on application. The J-1552 Starter may also be operated by float switch, pressure switch or thermostat.

Direct Current Starters listed are enclosed automatic starter with under voltage release and fused line switch.

Prices furnished on application for Direct Current Starters larger than 10 hp.

TABLE 154

CENTRIFUGAL PUMPS

Type "A" Side Suction Single-stage Centrifugal Pumps

Size of Suction	Size of Discharge, Inches	Economic Capacity in Gallons per Minute	Approximate Weight, Pounds	Standard Pulley, Inches		List Standard	List with Bronze Impeller	List for All Bronze Water End
				Diameter	Face			
2	1½	60	240	6	4	$96	$106	$190
3	2	100	260	6	4	100	110	200
4	3	250	500	8	5	150	165	340
5	4	450	1200	10	6	350	390	780
6	5	750	1400	14	8	380	420	810
8	6	1000	1600	18	10	420	470	850
10	8	1800	2300	20	10	650	700	1350
12	10	3000	2700	22	12	720	780	1600
14	12	4250	3000	24	14	800	875	2000

Standard Pump fitted with Steel Shaft and Cast Iron Impeller. Electric includes Flexible Couplings, Extended Sub-base and mounting of motor when delivered to our factory. Belted includes standard pulley and outboard bearing.

TYPE "E" SIDE SUCTION VOLUTE CASING SINGLE-STAGE CENTRIFUGAL PUMPS

Size of Suction	Size of Discharge, Inches	Economic Capacity in Gallons per Minute	Approximate Weight, Pounds	Standard Pulley, Inches		List Standard	List with Bronze Impeller	List for All Bronze Water End
				Diameter	Face			
2	1½	60	240	6	4	$96	$106	$190
3	2	100	260	6	4	100	110	200
4	3	250	500	8	5	150	165	340
5	4	450	700	10	6	180	200	375
6	5	750	1400	14	8	380	420	810
8	6	1000	1600	18	10	420	470	850
10	8	1800	1800	20	10	460	500	900

30 per cent discount from list price.

TABLE 155

OPERATING LABOR COSTS IN OIL ENGINE PLANTS

[Power, 1925]

Plant Number	Number of Units	Total Horse Power	Hours of Service per Day	Total Engine Hours per Year, All Units	Number of Attendants	Total Labor Cost, Dollars	Location
1	2	150	24	7,552	2	3000	North Dakota
2	3	175	24	11,315	3	3900	South Dakota
3	2	250	24	......	3	4800	Nebraska
4	1	50	10	3,650	1	2160	Nebraska
5	1	$37\frac{1}{2}$	14	4,750	1	1620	Iowa
6	2	200	24	8,800	3	3440	Iowa
7	2	100	24	8,395	2	2400	Nebraska
8	3	$237\frac{1}{2}$	24	10,030	3	4320	Nebraska
9	2	300	24	8,459	3	4320	Louisiana
10	1	100	12	4,380	1	1900	Ohio
11	1	50	24	8,010	1*	1200	California
13	2	300	24	8,760	3	4440	Michigan
14	2	250	24	6,200	3	2270	California
15	2	150	8	2,920	1	1200	Oklahoma
16	1	150	24	6,570	2	1740	Arkansas
17	3	250	24	14,620	3	4800	Texas
18	1	75	....	......	2	3000	Illinois
19	4	900	24	......	4	5760	Texas
20	3	350	24	......	2	2820	Texas
21	1	100	8	2,920	1	900	Kentucky
22	2	100	24	10,220	2	2400	Oklahoma
23	2	75	10	4,350	2	2500	Maine
24	3	600	24	5,760	3	2310	Tennessee
25	2	300	24	12,000	2	2010	Mississippi
26	2	150	24	5,923	2	3206	Texas
27	3	300	24	12,720	2	2300	Mississippi

* One man on each shift operates ice plant.

TABLE 156

AN ANALYSIS OF REPAIR COSTS IN THE OIL ENGINE PLANTS

Plant Number	Number of Units	Total Horse Power	Years of Service	Total Cost of Engine Repairs, Dollars	Extra Labor Cost, Dollars	Repairs per Unit Produced, Dollars
1	2	150	1½			
2	3	175	8	2506.00	....	0.00024
3	2	250	1			
4	1	50	2	0.84		
5	1	37½	3			
6	2	200	1			
7	2	100	6	1007.68	....	0.00016
8	3	237½	5	980.88	40.00	
9	2	300	4	150.00		
10	1	100	1½	65.00	50.00	
11	1	50	10	265.00		
12	1	100	4			
13	2	300	4	785.00	....	0.0005
14	2	250	8	368.00		
15	2	150	4	51.51	50.00	
16	1	150	1½			
17	3	250	10	6293.00*	....	0.0012
18	1	75	3	249.03		
19	4	900	2	605.15	....	0.00024
20	3	350	1			
21	1	100	1			
22	2	100	6	750.00		
23	2	75	6	1554.12		
24	3	600	1			
25	2	300	1	21.65		
26	2	150	4	527.67		
27	3	300	4	308.46	....	0.00027
28	2	400	2	414.90	523.98	
29	1	100	3	54.04		

* Crankshaft.

TABLE 157

OPERATING COSTS OF THE OIL-ENGINE DRIVEN ELECTRIC PLANTS

Plant Number	Load Factor, Per Cent		Kw.-Hr. Produced per Year	Operating Cost, Dollars	
	Running	Yearly		Total	Unit
1	$37\frac{1}{2}$	15	113,605	4,929	0.0434
2	42	18	165,050	6,787	.0411
6	82	41	430,000	7,320	.0170
7	48	23	120,000	4,080	.0340
13	52	36	403,000	8,497	.0210
17	58	37	481,800	10,309	.0214
19	....	36	1,271,400	15,224	.0120
24	57	21	650,000	10,236	.0157
27	38	23	365,000	5,648	.0155
				Average....	0.246

Plant Number	Total Investment, Dollars	Fixed Charge Cost, Dollars		Unit Fuel and Lubricating Oil Cost, Dollars	Unit Total Cost with Fixed Charges, Dollars
		Total	Unit		
1	19,800	1980	0.0174	0.016	0.0608
2	36,300	3630	.0220	.0147	.0631
6	30,000	3000	.0070	.0066	.0240
7	18,200	1820	.0152	.0140	.0490
13	41,000	4100	.0102	.0094	.0312
17	34,400	3440	.0071	.0079	.0285
19	74,400	7440	.0058	.0074	.0178
24	92,000	9200	.0141	.008	.0298
27	54,000	5400	.0148	.009	.0303
		Average....	0.0124	0.0103	

TABLE 158

OPERATING COSTS OF THE OIL ENGINE PUMPING PLANTS COMPARED WITH STEAM- AND MOTOR-DRIVEN PLANTS

Plant	Number of Pumps	Gallons Pumped, Millions	Static Head	Operating Cost Except Fuel and Lubricating Oil, Dollars	Cost per Million Gallon, Cent	Fuel and Lubricating Oil, Dollars	Fuel and Lubricating Oil per Million Gallons, Cents	Total Operating Costs per Million Gallons, Cents	Fuel and Lubricating Oil Cost Corrected for 150-Ft. Head, Cents	Total Operating Costs on Basis of 150-Ft. Head, Cents	Price of Fuel Oil Coal Current, Cents per Gallon
5	2	142	150	1,805	12.72	1,505	10.60	23.32	11.12	23.84	11.9
10	1	163.3	322	2,115	12.92	1,532	9.38	22.30	4.37	17.29	6.47
15	2	87.6	402	1,260	14.38	619	7.07	21.45	2.64	17.02	4.0
16	1	222.0	402	2,040	9.20	2,568	11.55	20.75	4.32	13.52	6.02
26	2	262.1	200	4,046	15.41	1,457	5.56	20.97	4.17	19.58	3.57
Steam Pumping Plant Costs											Dollars per Ton
Marinette	4	617.0	150	4,142	6.70	15,806	25.60	32.30	25.60	32.30	6.72
La Crosse	8	1213.0	240	18,780	15.45	19,419	16.00	31.45	10.00	25.45	6.89
Berlin	2	77.4	135	4,066	52.70	2,093	27.10	79.80	30.50	83.20	7.00
Washburn	2	155.8	250	2,840	18.25	6,186	39.75	58.00	23.80	42.05	9.33
Janesville	4	589.0	175	6,760	11.45	7,405	12.55	24.00	10.75	22.20	6.04
Kenosha	5	1670.0	162	17,491	10.45	19,816	11.85	22.30	10.97	21.42	7.60
South Milwaukee	6	317.0	210	11,536	36.50	11,847	37.40	73.90	26.70	63.20	7.60
Motor-driven Pumping Plant Costs							Current Costs				Cents per Kw.-Hr.
Menominee	2	170.0	240	3,361	19.80	5,633	33.10	52.90	20.70	40.50	1.25
Delavan	3	244.3	125	3,560	145.80	1,378	56.60	202.40	68.00	213.80	3.0
Columbus	2	466.5	135	767	16.50	1,573	34.80	51.30	38.70	55.20	3.5
Castleton	1	146.7	223	1,374	93.50	1,533	104.20	197.70	70.40	163.90	4.5
Neenah	3	177.0	110	3,642	20.60	8,246	46.50	67.10	39.40	60.00	
Wisconsin Rapids	4	214.0	195	2,619	12.20	5,034	23.50	35.70	18.05	30.25	2.44
Chippewa Falls	4	371.6	185	2,471	6.65	6,475	17.40	24.05	14.10	20.75	1.44
Lake Geneva	4	608.7	186	3,531	58.00	4,132	67.80	125.80	54.70	102.70	3.0
Stoughton	3	150.4	185	2,904	19.35	3,190	21.20	40.55	17.20	36.55	2.0
Waukesha	8	536.6	150	7,133	13.28	18,494	34.50	34.78	34.50	47.78	2.39

NOTE.—Motor and steam plant costs from Railroad Commission, State of Wisconsin.

TABLE 159

Depreciation of Ice Plants—G. E. Wells, A.S.R.E., 1920

	Per cent
Boilers	10
Pipe work	5
Brick building	2 to 3
Electric machinery	7
General machinery	7
Automatic equipment	20
Delivery equipment	10
Country ice houses	20
Office furniture and fixtures	10
Tools and implements	10

TABLE 160

Estimated Yearly Cost of Electric Plant Operation
[Power, 1926]

Output 30,000 tons annually. Based on actual operation, June to October, 1926

Investment:			
All electric equipment including installation and wiring cost			$25,000.00
Fixed charges:			
Interest, per cent		6	
Depreciation, per cent		4	
Insurance and taxes, per cent		2	
Total fixed charges		12	$3,000.00
Electricity:			
30,000 × 60 kw.-hr. per ton = 1,800,000 kw.-hr., at 1.14c.			20,500.00
Fuel for heating:			
20 tons coal at $6			120.00
Labor:			
1 Chief engineer at $45.00	$2340.00		
2 Night engineers at $38.00	3960.00		
1 Electrician at 30.00	1560.00		
3 Oilers and ice pullers at $25.00	3900.00		
Total			11,760.00
Supplies and repairs:			
25c. per ton × 30,000			7,500.00
Total			$42,880.00
Cost per ton			1.43

TABLE 161

ESTIMATED YEARLY COST OF OIL ENGINE PLANT OPERATION AND INCLUDING FIXED CHARGES

Output 30,000 Tons Annually

Investment:			
Two 200-hp. Diesel engines installed			$26,000.00
Two 200-hp. Diesel engines driving 140-kw. generators			33,000.00
200 hp. of electric motors, wiring and installation			7,000.00
			$66,000.00
Fixed charges:			
Interest, per cent		6	
Depreciation, per cent		8	
Insurance and taxes, per cent		2	
Total		16	$10,580.00
Fuel:			
6 gal. per ton × 30,000 = 180,000 gal. at $6\frac{1}{2}$ c.			11,700.00
Lubricating oil:			
1-10 gal. per ton × 30,000 = 3,000 gal. at 55c.			1,650.00
Labor:			
1 Chief engineer at $45.00	$2340.00		
2 Watch engineers at $38.00	3960.00		
1 Electrician at $30.00	1560.00		
4 Oilers and ice pullers at $25.00	5200.00		
Total			$13,060.00
Supplies and repairs:			
30 c. per ton × 30,000			9,000.00
Total			$45,990.00
Cost per ton			1.53

TABLE 162

OPERATION COSTS WITH AND WITHOUT ICE STORAGE

	200-Ton Plant		10,000-Ton Storage and 125-Ton Plant	
	Cost per ton	Total cost	Cost per ton	Total cost
General expense, including insurance, taxes, depreciation and interest on investment (14 per cent)	$1.68	$4,200	$1.55	$38,780
Plant labor	0.10	2,500	0.14	3,500
Tank-room labor	0.10	2,500	0.14	3,500
Day storage-room labor	0.06	1,500	0.10	2,500
Power	0.45	11,250	0.47	11,750
Ammonia	0.05	1,250	0.05	1,250
Water and other supplies	0.05	1,250	0.05	1,250
Maintenance	0.15	3,750	0.15	3,750
Salary of portion of crew during shutdown period	0.15	3,850 *	0.088	2,200†
Winter storage-room labor 10,000 tons	....	....	0.40	4,000
Winter storage-room refrigeration 10,000 tons	....	....	0.30	3,000
Totals	....	$69,850	....	$75,480
Average cost per ton	$2.79	....	$3.02	

Saving in favor of large daily capacity, $0.23 per ton.

* On basis of 5 months' operation; shutdown 7 months at $550 per month.

† On basis of 8 months' operation; shutdown 4 months at $550 per month.

CHAPTER XXI

SPECIFICATIONS

The object of specifications is to set down concisely, yet clearly and completely, all the requirements of a purchase. Either of two methods may be pursued. First the purchaser can state exactly what is desired in the apparatus or the machinery, and material entering into the construction and the types and the construction of the accessories and ask for no guarantee. Second, the purchaser can state the capacity required, or the results desired, and leave to the manufacturer the obligation to work out the details of the design and then specify a guarantee of performance. In the second case the reputation of the manufacturer is an important factor, as frequently the difficulty of making tests or of proving that the guarantee is lived up to, makes it of no value. As a rule it is much more satisfactory to specify standard equipment, for in such cases the price is usually considerably less than would be the case where new patterns and possibly special machinery must be provided in order to manufacture the special product.

Compressors.—The purchaser must needs make a choice of the type of compressor he desires. In America this choice is confined to ammonia and carbon dioxide as refrigerants. Although carbon dioxide is usually stated as a cold condensing water refrigerant, it is used to a very large extent on the merchant marine and even (to a slight extent) for stationary work in the southern part of the United States and Cuba. Notwithstanding that the power requirements are greater for carbon dioxide than for ammonia and that these increase with the temperature of the "liquid" from the condenser, yet the excess power for carbon dioxide over that for ammonia is only nominal. The real decision as to the kind of refrigerant should be made because of other factors such as safety, compactness, kind of load, etc.

The type of compressor and the kind of drive should be specified. The preferred types are: for 10 tons or less, the automatic; for 75 tons and less, the twin cylinder, vertical, single-acting machine; and for over 75 tons, either the horizontal double-acting type or the vertical single-acting machine with the open frame. There are decided advantages in the use of the enclosed compressor which in Great Britain, in particular,

are evidenced by the use of three or four cylinders on the same shaft with capacities of 200 tons and over.[1]

The rotative speed of the compressor should be specified. Speeds over 100 r.p.m. require careful machine design as regards the type of valve and the valve opening, stresses in the frame, reciprocating parts and bearings, and very particular attention to the lubrication. The old packing house design, steam engine driven, has become obsolete and the synchronous motor drive (where electric power is available) and the uniflow or poppet valve medium speed steam engine and oil engine driven compressor are replacing it. Experience has shown that the essential point to be considered in any case is the perfection of the lubrication which must be positive and sufficient in amount.[2] The use of stage compression should be specified for evaporating temperatures of 0 deg. F. and lower, especially where the condenser pressure is 150 lb. gage and over. Multiple effect compression has certain limited advantages in special cases and it will be worth consideration when two evaporating pressures are carried. Multiple effect compression can be used to advantage in carbonic compression for cooling the liquid refrigerant, and particularly when the water for the condenser is 70 deg. F. and higher.

Details on the design of the piping should be specified in full. The gas flow into and out of the compressor should be as direct as possible. Metallic packing causes much less friction than does fibre packing or leather cups, will not wear the rod as much and can be made gas-tight. By-pass valves and connections should be of ample size and convenient of operation. Means should be provided for suction from the atmosphere, when testing out under air pressure, and discharge to the atmosphere, when pumping a vacuum, without having to remove fittings or flange connections. The oil and scale trap should be of ample size and should be designed for convenience as to the removal of oil and scale. For carbonic refrigeration it is well to specify that the scale trap, oil separator, etc., should be welded. Mention should be made of the desired size of the suction and condenser pressure gages, and the size and the location of the safety pop valve, whether discharging to the atmosphere or to the low-pressure side of the system.

Condensers.—Ammonia condensers should be erected with purge and pump-out connections. The amount of water available, and its temperature, should be mentioned, and the kind of condenser should be specified as well as the amount of the condensing surface, which in all cases should be liberal.[3] The modern condensers for ammonia are the

[1] See Chapter II.

[2] See construction details in Chapter II.

[3] See Chapter VI.

common atmospheric, the drip type, the double pipe, and the shell and tube. Compactness and moderate efficiency have brought the shell and tube condenser into popularity, especially in the larger sizes. Of the pipe condensers the common and the drip type are the popular atmospheric condensers in the order named, and the double pipe is used where the water is reasonably clear and particularly for automatic and other small installations. In carbonic work the welded double pipe and the modified submerged coil type are preferred. The number of pipes per stand should be mentioned, although it is doubtful whether more than twelve pipes are economical. The number and the spacing of the stands, the piping arrangement of the headers and the liquid drain, the kind of fittings, the water distributor and valves should be specified. If the spacing of the pipes for the atmospheric condenser is $4\frac{1}{2}$ in. or over, splash strips should be provided. As some manufacturers make use of full weight steel pipe, whereas others use extra heavy steel or wrought iron, the kind of pipe should be mentioned.

The Liquid Receiver should be specified by the diameter and length, and the thickness of the shell and of the heads should be given. The liquid receiver should be tested before shipment with a 500 lb. air pressure under water. The volume of the receiver should bear a relationship to the amount of the liquid in the system (see Chapter X), and in the smaller sizes the ability to store the entire amount of the charge in the receiver is very useful. The gage glasses should be protected in an approved manner, and a safety valve to prevent an excess pressure in the receiver should be required.[4]

Fittings.—Fittings may be screwed or flanged; of malleable iron, semi-steel or drop forged. The extensive use of welding has stimulated the use of steel flanges particularly, as well as steel fittings and valves because of the ease in welding, especially in the case of carbonic refrigeration. As a rule the screw fitting is confined to the smaller pipe sizes, whereas the larger sized pipe uses the oval, the square, or the round flange, depending on the size. The accepted standard flange has the tongue and groove, using a lead, rubber or asbestos gasket. In carbonic refrigeration it has been found that the only tight flange is that made with the end of the pipe and a hard gasket. Care should be taken in the specifications so as to include the kind of fittings desired.

Practically all ammonia valves are designed so as to permit packing of the valve stem while the valve is under pressure. Either the soft or the hard seat valve may be specified, and both have relative advantages. The soft seat valve will be tight until the lead is squeezed out and

[4] See details on liquid receivers in Chapter IV.

reseating becomes necessary. The scale trap should be mentioned, giving details for cleaning.

Piping.—Steel pipe now is used almost exclusively in ammonia refrigeration, although wrought iron is still specified. There is no standard weight of pipe in the high-pressure side, as a number of manufacturers use full weight pipe. Likewise both full weight and extra heavy pipe is specified for the direct expansion piping in ice tanks. It is usual to specify butt welded pipe in 2-in. sizes and under, and lap welded pipe in sizes over 2 in.

In ammonia direct expansion piping fittings are dispensed with as much as possible. The "hair pin" coil is now used in many cases where continuous welding is not used. Detail drawing should be made of all piping, both on the high and the low-pressure side.

Ice making tanks should be given the exact dimensions desired, including the thickness of the plate, the bulkheads, partitions, scarfing of the seams, reinforcing, the boxes and the drains. If a brine cooler is used, the diameter, length and the amount of surface desired should be given. The tubes are frequently 2-in. o.d. of No. 12 charcoal iron and are expanded into the tube sheet, which is approximately $\frac{1}{2}$ in. thick. A single pass is used for ice tanks and 4, or even as many as 8, passes for general brine cooling. A pump out, oil drain and liquid gage connection should be provided.

The size and the number of cans, the amount of agitation of the brine and the details of the air agitation in the cans should be specified (Chapter XIII). The method of water softening, the use of cooling towers or of sprays, the kind of crane for the cans and the number of cans to be lifted at a time, the use of baskets, the manner of thawing, etc., should be specified and not left to the manufacturer.

Brine.—Calcium chloride brine is specified for most large work not using brine sprays. Such brine is usually best when it is free from magnesium, and it should be of such a concentration as will be safe from freezing at the lowest operating temperature.[5]

The following tables give some idea of the detailed weights and sizes of standard equipment.

[5] See Chapter VIII.

TABLE 163

LOW-PRESSURE FREEZING SYSTEM APPARATUS

Tons of Ice	Number of Cans	Air Blowers								Core Pumping Set				
		Number of blowers	Size of pipe, inches	C.f.m.	R.p.m.	Horse power motor required	Size of blower pulley, inches	Belted, weight, pounds,	Maximum r.p.m. motor	Number	Size of pump	Belted, weight, pounds	Motor-driven, weight, pounds	Horse power motor required
1	16	1	2½	8	520	.75	7×1.5	90	1200	1	1¼	200	260	2
2	30	1	2½	15	635	.75	7×1.5	90	1200	1	1¼	200	260	2
3	42	1	2½	21	600	1.5	8×2	200	1200	1	1¼	200	260	2
4	56	1	2½	28	670	1.5	8×2	200	1200	1	1¼	200	260	2
5	70	1	2½	35	740	2.0	8×2	200	1800	1	1¼	200	260	2
6	84	1	2½	42	810	2.0	8×2	200	1800	1	1¼	200	260	2
8	112	1	2½	56	700	3.0	10×2.5	300	1800	1	1¼	200	260	2
10	144	1	2½	72	800	3.0	10×2.5	300	1800	1	1¼	200	260	2
12	168	1	3	84	505	5.0	12×3.0	400	1800	1	1¼	200	260	2
15	210	1	3	105	550	5.0	12×3.0	400	1200	1	1¼	200	260	2
20	280	1	3	140	660	5.0	12×3.0	400	1200	1	1¼	200	260	2
25	360	1	3	180	790	5.0	12×3.0	400	1200	1	1¼	200	260	2
30	420	1	4	210	460	7.5	16×3.0	730	1200	1	1¼	200	260	2
35	484	1	4	242	505	7.5	16×3.0	730	1200	1	1¼	200	260	2
40	560	1	4	280	565	7.5	16×3.0	730	1200	1	1¼	200	260	2
50	720	1	4	360	680	7.5	16×3.0	730	1200	1	1¼	200	260	2
60	840	2	4	420	460	7.5	16×3.0	1460	1200	1	1¼	200	260	2
80	1120	2	4	560	565	7.5	16×3.0	1460	1200	1	1¼	200	260	2
100	1440	2	4	720	680	7.5	16×3.0	1460	1200	1	1¼	200	260	2
120	1680	2	6	840	500	10.0	20×4.0	3000	1200	1	1¼	200	260	2
150	2160	2	6	1080	550	15.0	20×4.0	3000	1200	1	1¼	200	260	2
200	2880	3	6	1440	550	15.0	20×4.0	4500	1200	2	1¼	400	520	2

Core Pumping Set consists of centrifugal pump, hydraulic ejector and water tank and connections.
Blowers operate at 3 lb. gage pr. Use 1800 r.p.m. motors for motor-driven core pumping sets.

TABLE 164

6-ton Distilled Water Ice Making Plant—Flooded (300-Lb. Cans)

Items	Weight, Pounds	List Price
1 Refrigerating machine, $7\frac{1}{2}\times10\frac{1}{2}\times6$ single column	11,000	1550
Painting machine at the factory	....	27
1 Double pipe ammonia condenser, 8 pipes high, 30 ft. long, $1\frac{1}{4}\times2$; black, full weight, wrought iron pipe	1,950	155
1 Liquid receiver, welded, 12 in. dia by 12 ft. long	600	72
1 Oil separator, welded, 12 in. dia. by 3 ft. long	160	25
Ammonia pipe connections, extra heavy discharge, full weight suction	1,000	191
1 Freezing tank, 23 ft. by 11 ft. 4 in by 4 ft., $\frac{1}{4}$ in. steel	5,900	300
1080 ft. $1\frac{1}{4}$ in. full weight black steel tank coils with stands and headers	4,750	272
1 Accumulator, welded, 16 in. dia. by 4 ft. 9 in. long	230	29
72 Oak covers and frames with can guides	2,880	97
72 Freezing cans, 300 lb., $11\frac{1}{2}\times22\frac{1}{2}\times45$	5,040	364
1 Propeller agitator, 15 in. diameter	410	38
Bulkhead and partitions for tank	400	18
1 Crane and hand hoist, 300 lb. single can	1,220	94
1 Can dump 300 lb. single iron type	450	40
1 Can filler and 20 ft. 5-ply hose	40	31
1 Exhaust steam oil separator, welded, 16 in. by 4 ft. long	210	36
1 Shell steam condenser, 9 ft. 9 in. long, No. 10, iron, galvanized and trough	580	61
1 Distilled water pump, steam driven, 4 gal. per min	100	42
1 Float tank, 15 in. dia. by 24 in. long and float control	115	22
1 Reboiler 7 in.$\times$16 in.$\times$18 in. and steam coil, painted	380	50
1 Automatic regulator, $\frac{3}{4}$ in. connection	75	14
1 Double pipe distilled water cooler, 6 pipes high, 10 ft. long, $1\frac{1}{4}\times2$ in. full weight galvanized	480	50
2 Distilled water filters, 24 in. dia. by 5 ft. long with charcoal to charge	1,230	108
1 Distilled water storage tank, 7 ft.$\times$4 ft.$\times$3 ft. 6 in. $\frac{3}{16}$ in. iron, with 68 ft. extra heavy 2 in. pipe	1,560	128
Distilled water pipe connections, galvanized	300	146
1 Horizontal return tubular boiler complete, 54 in. dia.$\times$14 ft. long, 60 hp.	15,665	993
1 Boiler feed pump, steam driven, 9 gal. per min	180	63
1 Feed water heater, 60 hp. vertical closed type, brass tubes	550	103
Steam and exhaust pipe connections	600	218
$4\frac{1}{2}$ Tons of salt	9,000	58
300 Lb. ammonia	300	41
Engineer	....	585
Usual labor	......	585
Union labor	......	1170
$5\frac{1}{4}$ Tons calcium	10,500	133

TABLE 165

$1\frac{1}{4}$-In. and 2-In. Double Pipe Ammonia Condensers, 10 and 20 Ft. Long, Specifications, Weights and Prices

Pipes, High, Inches	Effective Surface, Square feet	Height Over Top Pipe, Inches	Height Over All, Inches	Full Weight of Pipe, Pounds	Extra Heavy Pipe, Pounds	Add for Soldered Joints
10-Ft. Coils						
6	21	32	47	750	840	17
8	28	40	55	1000	1120	22
10	35	48	63	1250	1400	28
12	42	56	71	1400	1625	33
14	49	64	79	1700	1900	39
20-Ft. Coils						
4	31.5	24	39	720	900	11
6	47	32	47	1080	1320	17
8	62.5	40	55	1440	1760	22
10	78	48	63	1800	2200	28
12	94	56	71	2100	2620	33

Gas, liquid and purge valves and 2 pr. flanges for water connections included. No headers nor water valves included.

TABLE 166

Condenser Pans

Ammonia Condenser Pans			Flask Steam Condenser Pans			Steam Condenser and Distilled Water Cooler Pans		
Number of coils	Size of pan	Weight, pounds	Number of flasks	Size of pan	Weight, pounds	Number of coils	Size of pan	Weight, pounds
	Ft. Ft. In.			Ft. Ft. In.			Ft. Ft. In.	
1	14× 4×6	600	1	12× 4 ×6	520	1	12× 4×6	520
1	24× 4×6	1000	1	15× 4 ×6	640	1	22× 4×6	920
2	24× 6×6	1400	1	18× 4 ×6	760	2	22× 6×6	1300
3	24× 7×6	1600	1	22× 4 ×6	920	3	22× 8×6	1925
4	24× 9×6	2000	2	12× 6 6×6	780	4	22×10×6	2100
5	24×10×6	2200	2	15× 6 6×6	960	5	22×12×6	2550
6	24×12×6	2700	2	18× 6 6×6	1200	6	22×14×6	3000
7	24×13×6	3000	2	22× 6 6×6	1400	7	22×16×6	3400
8	24×15×6	3400	3	15× 9 ×6	1300	8	22×18×6	3800
9	24×16×6	3800	3	18× 9 ×6	1600	9	22×20×6	4200
10	24×18×6	4200	3	22× 9 ×6	1900	10	22×22×6	4600
11	24×19×6	4400	4	22×11 6×6	2450	11	22×24×6	5000
12	24×21×6	4800	5	22×14 ×6	3000	12	22×26×6	5400
13	24×22×6	5000	6	22×16 6×6	3500	13	22×28×6	5800
14	24×24×6	5400	7	22×19 ×6	4000	14	22×30×6	6200
15	24×25×6	5600	9	22×24 ×6	5000	15	22×32×6	6600
16	24×27×6	6000	11	22×29 ×6	5900	16	22×34×6	7000
18	24×30×6	6600	14	22×36 6×6	7200	18	22×38×6	7400

Note.—All above pans made of $\frac{3}{16}$-in. material. Ammonia condenser pan sizes given for coils on 18-in. centers.

Sizes above heavy line are built up at factory. Sizes below are shipped knocked down.

TABLE 167

ENCLOSED TYPE MACHINERY AND HIGH SIDES

	Weight, Pounds
1 10-in.×10-in., two-cylinder, belt-driven machine	11,400
3 Double pipe ammonia condensers, 10 P.H.×20 ft. long, $1\frac{1}{4}$ in. and 2-in. F. W. steel pipe	5,400
Gas, liquid and water headers for condensers	324
1 Ammonia oil separator, 16 in.×4 ft.	370
1 Ammonia receiver, 20 in.×10 ft.	850
Ammonia connections:	
60 ft., $2\frac{1}{2}$ in. E. H. steel pipe	
5 $2\frac{1}{2}$ in. ammonia ells, C. F. B. & G.	
2 Pr. $2\frac{1}{2}$ in. ammonia flanges B. & G.	
20 ft., 3 in. F. W. steel pipe	1,250
3 3-in. ammonia ells, C. F. B. & G.	
20 ft., 1 in. E. H. pipe	
2 Pr. 1 in. ammonia flanges B. & G., gage connections	
Total	19,594
1 10-in.×10-in., two-cylinder machine arranged for horizontal uniflow engine drive	9,000
1 10-in.×10-in., two-cylinder machine arranged for engine type synchronous motor drive. Includes wheel, shaft extension, coupling and outboard bearing. Shaft swell permissible. No exciter pulley included.	12,360
Add to above for pulley when belted exciter is used	500
Deduct wheel at rate of 12 cents per pound list.	
1 12-in.×12-in., two-cylinder, belt-driven machine	21,300
4 Double pipe ammonia condensers, 12 P.H.×20 ft. long, $1\frac{1}{4}$-in. and 2-in. F. W. steel pipe	8,400
Gas, liquid and water headers for condensers	448
1 Ammonia oil separator, 20 in.×5 ft.	600
1 Ammonia receiver, 20 in.×16 ft.	1,200
Ammonia connections:	
60 ft., $3\frac{1}{2}$ in. E. H. steel pipe	
5 $3\frac{1}{2}$ in. ammonia ells, C. F. B. & G.	
2 Pr. $3\frac{1}{2}$ in. ammonia flanges B. & G.	
40 ft., 4 in. F. W. steel pipe	2,200
3 4 in. ammonia ells, C. F. B. & G.	
40 ft. $1\frac{1}{4}$ in. E. H. pipe	
2 Pr. $1\frac{1}{4}$ in. ammonia flanges, B. & G., gage connections	
Total	34,148
1 12-in.×12-in., two-cylinder machine arranged for horizontal uniflow engine drive	15,300
1 12-in.×12-in., two-cylinder machine arranged for engine type synchronous motor drive. Included wheel, shaft extension, coupling and outboard bearing. Shaft swell permissible. No exciter pulley included.	23,050
Add to above for pulley when belted exciter is used	600

TABLE 168

HORIZONTAL UNIFLOW ENGINES—SPECIFICATIONS

Steam Cylinder, Inches		Flywheel			Main Bearing, Inches		Crank Pin, Inches		Cross Head Pin, Inches		Connecting Rod, Center to Center, in Inches	Piston Rod Diameter, Inches	Pipe Connections, Inches	
Diameter	Stroke	Diameter		Weight, pounds	Diameter	Length	Diameter	Length	Diameter	Length			Steam	Exhaust
		Ft.	In.											
12	12	6	0	2,500	6	12	$4\frac{1}{2}$	4	3	5	40	$2\frac{5}{8}$	$3\frac{1}{2}$	$4\frac{1}{2}$
12	14	6	0	2,500	6	12	$4\frac{1}{2}$	4	3	5	40	$2\frac{5}{8}$	$3\frac{1}{2}$	$4\frac{1}{2}$
14	14	7	0	3,800	7	14	5	5	$3\frac{1}{2}$	$5\frac{1}{2}$	44	3	4	5
14	16	7	0	3,800	7	14	5	5	$3\frac{1}{2}$	$5\frac{1}{2}$	44	3	4	5
16	16	8	0	5,000	8	16	6	6	4	$6\frac{1}{2}$	48	$3\frac{1}{2}$	5	6
16	18	8	0	5,000	8	16	6	6	4	$6\frac{1}{2}$	48	$3\frac{1}{2}$	5	6
18	18	8	6	6,500	9	18	7	6	$4\frac{1}{2}$	$7\frac{1}{2}$	54	$3\frac{7}{8}$	6	7
18	20	8	6	6,500	9	18	7	6	$4\frac{1}{2}$	$7\frac{1}{2}$	54	$3\frac{7}{8}$	6	7
20	20	9	6	8,000	10	20	8	7	5	9	60	$4\frac{1}{4}$	7	8
20	22	9	6	8,000	10	20	8	7	5	9	60	$4\frac{1}{4}$	7	8
22	22	9	6	10,000	11	22	9	8	6	$9\frac{1}{2}$	66	$4\frac{3}{4}$	7	9
24	22	9	6	12,000	11	22	9	8	6	$9\frac{1}{2}$	$60\frac{1}{2}$	$5\frac{1}{8}$		
22	24	9	6	10,000	11	22	9	8	6	$9\frac{1}{2}$	66	$4\frac{3}{4}$	7	9
24	24	9	6	13,500	12	24								
24	26	9	6	13,500	12	24								
26	26	10	0	15,000	13	26								
26	28	10	0	15,000	13	26								
27	28	10	0	16,500	$13\frac{1}{2}$	27	....	....	....	....	$82\frac{1}{2}$			
27	30	10	0	16,500	$13\frac{1}{2}$	27	11	10	$7\frac{1}{2}$	11	$82\frac{1}{2}$	$5\frac{1}{8}$	10	12

Engines 24 in. × 24 in. and larger are equipped with tail rod. When engine room space will not permit use of tail rod, consult Home Office for engine recommendations. When engines are to be used with H. D. A. long stroke compressors the disengaging type valve gear with fly-ball governor will be used.

TABLE 169

FREEZING TANKS, BULKHEADS AND PARTITIONS COIL TANKS, 300 LB. STANDARD CANS

Tons of Ice, Basis of 16 Cans per ton	Number of Cans	Standard Tank Sizes Using Horizontal Agitators. For Vertical Agitators Add 7 In. to Tank Length. Tanks 4 Ft. 0 In. Deep. Material ¼-In. Tank Steel.				Weight per Square Foot	Weight, Pounds	Bulkheads and Partitions	
		Length		Width				Number of agitators	Weight, pounds
		Ft.	In.	Ft.	In.				
1	16	11	0	5	10	12	2,390	1	200
	15	13	0½	4	7½	12	2,440	1	220
2	30	15	1	7	0½	12	3,410	1	250
	30	13	0½	8	3	12	3,360	1	240
3	42	15	1	9	5½	12	4,080	1	270
	42	17	1½	8	3	12	4,150	1	280
4	56	17	1½	10	3	11½	4,660	1	290
	54	21	2½	8	3	11½	4,730	1	300
5	72	19	2	11	10½	11½	5,460	1	340
	70	23	3	9	5½	11½	5,560	1	350
5½	80	19	2	13	1	11½	5,860	1	360
	80	23	3	10	8	11½	5,980	1	370
6	99	21	2½	14	3½	11½	6,800	1	380
	99	25	3½	11	10½	11½	6,830	1	390
7½	120	23	3	15	6	11¼	7,550	1	400
	120	27	4	13	1	11¼	7,690	1	420
9	144	27	4	15	6	11¼	8,630	1	440
	144	35	6	11	10½	11¼	9,000	1	480
11	182	29	4½	17	11	11¼	10,240	1	500
	180	33	5½	15	6	11¼	10,260	1	520
15	240	33	5½	20	4	11¼	12,630	1	550
	238	37	6½	17	11	11¼	12,850	1	600
19	306	37	6½	22	9	11¼	15,370	1	650
	304	42	3½	20	4	11¼	15,420	1	775
22	360	39	7	25	2	11	17,000	2	1000
	360	44	4	22	9	11	17,090	1	900
26	418	42	3½	27	7	11	19,100	2	1150
	420	46	4½	25	2	11	19,240	2	1200
30	484	48	5	27	7	11	21,430	2	1300
	480	52	6	25	2	11	21,490	2	1350
37½	600	54	6½	30	0	11	25,500	2	1400
	600	64	9	25	2	11	25,890	2	1450
45	720	64	9	30	0	11	29,830	2	1500
	720	77	0	25	2	11	30,340	2	1550

For larger tonnages than those given use two or more smaller tanks of proper size.

TABLE 169*a*

FREEZING TANKS, BULKHEADS AND PARTITIONS COIL TANKS, 400 LB. STANDARD CANS

Tons of Ice, Basis of 12 Cans per ton	Number of Cans	Standard Tank Sizes Using Horizontal Agitators. For Vertical Agitators Add 7 In. to Tank Length. Tanks 5 Ft. 1 In. Deep. Material ¼-in. Tank Steel.				Weight per Square Foot	Weight, Pounds	Bulkheads and Partitions	
		Length		Width				Number of agitators	Weight, pounds
		Ft.	In.	Ft.	In.				
5	56	17	1½	10	8	11½	5,360	1	370
4	54	21	2½	8	3	11½	5,480	1	390
5	63	17	1½	11	10½	11½	5,760	1	390
5	63	21	2½	9	5½	11½	5,790	1	420
7	80	19	2	13	1	11¼	6,500	1	470
7	80	23	3	10	8	11¼	6,670	1	470
8	90	21	2½	13	1	11¼	7,060	1	500
7	88	25	3½	10	8	11¼	7,160	1	520
9	108	21	2½	15	6	11¼	7,920	1	550
9	108	27	4	11	10½	11¼	8,170	1	580
11	132	27	4	14	3½	11¼	9,180	1	600
11	135	33	5½	11	10½	11¼	9,710	1	700
15	182	29	4½	17	11	11¼	11,480	2	950
15	180	33	5½	15	6	11¼	11,570	1	850
20	224	31	5	20	4	11¼	13,220	2	1000
20	224	35	6	17	11	11	13,590	2	1100
20	270	33	5½	22	9	11	14,810	2	1200
20	272	37	6½	20	4	11	15,200	2	1350
25	320	35	6	25	2	11	17,000	2	1425
25	320	44	4	20	4	11	17,290	2	1550
30	360	39	7	25	2	11	18,590	2	1650
30	360	44	4	22	9	11	18,770	2	1725
40	460	50	5½	25	2	11	22,530	2	1800
40	450	54	6½	22	9	11	22,420	2	1875
45	540	58	7½	25	2	11	25,730	2	1950
45	540	64	9	22	9	11	26,120	2	2000

For larger tonnages than those given use two or more smaller tanks of proper size.

TABLE 170

DISTILLED WATER ICE MAKING APPARATUS

D. P. Distilled Water Cooler						Pipe Connections, Weight, Pounds
Tons of ice	Number of coils	Pipes high	Length Ft.	Length In.	Weight, pounds	
1	1	2	7	0	155	130
2	1	2	10	0	225	140
3	1	3	10	0	315	150
4	1	4	10	0	385	150
5	1	6	10	0	585	300
6	1	6	10	0	585	300
8	1	7	10	0	665	300
10	1	8	10	0	750	600
12	1	8	10	0	750	600
15	1	6	20	0	1,000	650
20	1	8	20	0	1,300	670
25	2	6	20	0	2,000	680
30	2	6	20	0	2,000	700
35	2	8	20	0	2,600	800
40	2	8	20	0	2,600	900
50	3	8	20	0	3,900	950
60	3	8	20	0	3,900	1000
75	4	8	20	0	5,200	1200
100	5	8	20	0	6,500	1500
125	7	8	20	0	9,100	2000
150	8	8	20	0	10,400	2400
200	10	8	20	0	13,000	3000

Distilled Water Filters

Tons of Ice	Number of Filters	Size of Filters, Inches	Weight, Pounds
1	1	24×60	6000
2 to 30	2	24×60	1200
35 to 40	3	24×60	1700
50 to 60	4	24×60	2300
75	5	24×60	2800
100	6	24×60	3350
125	7	24×60	3900
150	8	24×60	4475
200	10	24×60	5575

Distilled water coolers are made of $1\frac{1}{4}$-in. and 2-in. F. W. Galvanized pipe. Weights include cooler only. No valves nor headers.

Pipe connections include all necessary distilled water piping and fittings.

TABLE 170—*Continued*

Tons of ice	Flask Steam Condensers				Tons of ice	Flask Steam Condensers			
	Number of coils	Length		Weight, pounds		Number of coils	Length		Weight, pounds
		Ft.	In.				Ft.	In.	
1	1	9	9	670	30	3	11	9	2,390
2	1	9	9	670	35	3	15	6	3,170
3	1	9	9	670	40	3	19	6	3,915
4	1	9	9	670	50	4	19	6	5,120
5	1	9	9	670	60	5	19	6	6,530
6	1	9	9	670	75	6	19	6	7,840
8	1	9	9	670	100	7	19	6	9,150
10	1	11	9	750	125	9	19	6	11,750
12	1	15	6	1010	150	11	19	6	14,370
15	2	9	9	1420	200	14	19	6	18,200
20	2	11	9	1580					
25	2	15	6	2115					

Tons of ice	Reboilers		Tons of ice	Reboilers	
	Size of tanks	Weight, pounds		Size of tanks	Weight, pounds
	Ft. In. In. Ft.			Ft. In. In. Ft.	
1 thru 6	7 0 ×16×18	440	35 and 40	17 $\frac{5}{8}$×30×18	1350
8 thru 12	7 0 ×30×18	630	50 and 60	20 6 ×30×18	1550
15 and 20	10 $3\frac{5}{8}$×30×18	900	75 and 100	23 $9\frac{5}{8}$×30×18	1820
25 and 30	13 9 ×30×18	1100	125 thru 200	27 0 ×30×18	2170

Distilled Water Refrigerators

Tons of Ice	Connections, Inches	Weight, Pounds	Tons of Ice	Connections, Inches	Weight, Pounds
1 to 12	$\frac{3}{4}$	80	50 to 60	2	91
15 to 20	1	83	75 to 100	$2\frac{1}{2}$	100
25 to 30	$1\frac{1}{4}$	85	125 to 200	3	108
35 to 40	$1\frac{1}{2}$	87			

Tons of Ice	Float Tanks		Tons of Ice	Float Tanks	
	Size of tank, inches	Weight, pounds		Size of tank, inches	Weight, pounds
1 to 6	15×24	115	8 to 200	20×30	205

TABLE 170—*Continued*

Distilled Water Pumps

Size of Cylinder, Inches		Stroke, Inches	Pipe Connections, Inches				Weight, Pounds
Steam	Water		Steam	Exhaust	Suction	Discharge	
3	2	$3\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{4}$	$1\frac{1}{4}$	1	125
$3\frac{1}{2}$	$2\frac{1}{4}$	4	$\frac{1}{2}$	$\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	200

Steam condensers are made of 10-lb. galvanized steel and are furnished complete with stands, sprinkler, troughs and steam header.

Reboilers are made of $\frac{1}{8}$-in. steel with 16-lb. steel cover and vent pipes. Steam coil and overflow and water connections included. All painted.

Float tanks and pumps used as substitute for regulators when gravity feed to water storage tank not possible. Float tanks are made of $\frac{1}{8}$-in. steel and have float control.

TABLE 171

SPECIFICATIONS ENCLOSED TYPE V. S A. AMMONIA COMPRESSORS

Number of Cylinders	Cylinders, Inches		Fly Wheel			Crank Shaft, Inches		Crank Pins, Inches	
	Bore	Stroke	Diameter, inches	Face, inches	Weight, pounds	Diameter	Length of bearings	Diameter	Length
1	3	3	26	$3\frac{1}{2}$	180	$1\frac{3}{4}$	5	$1\frac{1}{2}$	$1\frac{3}{4}$
2	3	3	26	$3\frac{1}{2}$	180	$1\frac{3}{4}$	5	$1\frac{1}{2}$	$1\frac{3}{4}$
1	4	4	30	$4\frac{1}{2}$	300	2	5	2	$2\frac{3}{8}$
2	4	4	30	$4\frac{1}{2}$	300	2	5	2	$2\frac{3}{8}$
2	5	5	36	$6\frac{1}{2}$	600	$2\frac{1}{2}$	6	$2\frac{1}{2}$	$2\frac{3}{4}$
2	6	6	42	$7\frac{1}{2}$	600	3	7	3	$3\frac{1}{4}$
2	7	7	48	$8\frac{1}{2}$	1000	$3\frac{1}{2}$	8	$3\frac{1}{2}$	$3\frac{3}{4}$
2	8	8	50	$8\frac{1}{2}$	1200	4	9	4	$4\frac{3}{4}$
2	9	9	60	$10\frac{1}{2}$	1500	$4\frac{1}{2}$	$10\frac{1}{4}$	$4\frac{1}{2}$	$5\frac{1}{4}$
2	10	10	72	$12\frac{1}{2}$	1800	5	$11\frac{1}{4}$	5	$5\frac{3}{4}$
2	12	12	86	$14\frac{1}{2}$	3250	7	14	7	6

Number of Cylinders	Piston Pins, Inches		Connection Rods, Inches		Pipe Connections, Inches					
	Diameter	Length	Length, center to center	Diameter of bolts	Ammonia suction	Ammonia discharge	Water jacket inlet	Water jacket outlet	Steam	Exhaust
1	$\frac{7}{8}$	$1\frac{1}{2}$	$7\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{3}{8}$		
2	$\frac{7}{8}$	$1\frac{1}{2}$	$7\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{2}$		
1	$1\frac{1}{8}$	2	10	$\frac{1}{2}$	1	1	$\frac{3}{4}$	$\frac{3}{4}$	1	$1\frac{1}{4}$
2	$1\frac{1}{8}$	2	10	$\frac{1}{2}$	1	1	1	1	1	$1\frac{1}{2}$
2	$1\frac{1}{4}$	$2\frac{3}{4}$	$12\frac{1}{2}$	$\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{1}{4}$	1	1	$1\frac{1}{2}$	2
2	$1\frac{1}{2}$	$3\frac{1}{4}$	15	$\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	2
2	$1\frac{3}{4}$	$3\frac{3}{4}$	$17\frac{1}{2}$	$\frac{7}{8}$	2	$1\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	2	$2\frac{1}{2}$
2	2	$4\frac{3}{8}$	20	1	$2\frac{1}{2}$	2	$1\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{1}{2}$	3
2	$2\frac{1}{4}$	$4\frac{7}{8}$	$22\frac{1}{2}$	$1\frac{1}{8}$	$2\frac{1}{2}$	2	$1\frac{1}{2}$	$1\frac{1}{2}$	3	5
2	$2\frac{1}{2}$	$5\frac{3}{8}$	$25\frac{1}{2}$	$1\frac{1}{4}$	3	$2\frac{1}{2}$	2	2		
2	$3\frac{1}{2}$	$6\frac{1}{2}$	30	......	4	$3\frac{1}{2}$	2	2		

TABLE 172

ENCLOSED TYPE MACHINES AND HIGH SIDES

Items
1 4-in.×4-in., two cylinder B. D. machine. 1 D. P. ammonia condenser, 4 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel. 1 Ammonia oil separator, 8 in.×20 in. long. 1 Ammonia receiver, 6 in. diameter×8 ft. long. Ammonia connections: 15 ft. of 1 in. E. H. steel pipe. 7 1-in. ammonia ells, screwed. 2 Pr. 1-in. ammonia flanges B. & G. Gage connections. Machine arranged for direct engine drive.
1 5-in.×5-in., two cylinder B. D. machine. 1 D. P. ammonia condenser, 6 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel. 1 Ammonia oil separator, 8 in.×20 in. long. 1 Ammonia receiver, 8 in.×8 ft. long. Ammonia connections: 20 ft., 1¼ in. E. H. steel pipe. 7 1¼-in. ammonia ells screwed. 2 Pr. 1¼-in. ammonia flanges, B. & G. Gage connections. Machine arranged for direct engine drive.
1 6-in.×6-in., two cylinder B. D. machine. 1 D. P. ammonia condenser, 8 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel. 1 Ammonia oil separator, 12 in.×36 in. 1 Ammonia receiver, 8 in.×8 ft. long. Ammonia connections: 20 ft., 1½-in. E. H. steel pipe. 7 1½-in. ammonia ells screwed. 2 Pr. 1½-in. ammonia flanges, B. & G. Gage connections. Machine arranged for direct engine drive.
1 7-in.×7-in., two cylinder B. D. machine. 1 D. P. ammonia condenser, 10 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel. 1 Ammonia oil separator, 12 in.×36 in. 1 Ammonia receiver, 8-in.×18 ft. Ammonia connections: 25 ft., 1½-in. E. H. steel pipe. 5 1½-in. ammonia ells screwed. 2 2-in. ammonia ells screwed. 2 Pr. 1½-in. ammonia flanges, B. & G. Gage connections. Machine arranged for direct engine drive.

NOTE.—The ammonia condensers listed for all sizes Enclosed Type Machine H. S. are based on average conditions. If the conditions are extreme, determine proper size condenser from capacity sheets. The gage connections furnished with 4 in.×4 in. thru 9 in.×9 in enclosed.

Type Machine High Side consists of the following: Gages L. and H. pressure 20 ft. ¼ in. E. H. steel pipe, 2 ¼ in. angle valves, screwed.
2 ¼ in. ammonia ells, screwed. 4 Prs. ¼ in. ammonia flanges B. & G.

TABLE 172—*Continued*

Items
1 8-in.×8-in., two cylinder B. D. machine. 2 D. P. ammonia condensers, 8 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel. Gas and liquid headers. 1 Ammonia oil separator, 12 in.×36 in. 1 Ammonia receiver, 16 in.×7 ft. Ammonia connections: 30 ft., 2 in. E. H. steel pipe. 5 2 in. ammonia ells C. F. B. & G. 2 2½ in. ammonia ells C. F. B. & G. 2 Pr. 2-in. ammonia flanges, B. & G. 10 ft., ½ in. E. H. pipe. 1 ½-in. ammonia screw tee. 1 Pr. ½-in. ammonia flanges B. & G. Gage connections. Machine arranged for direct engine drive.
1 9-in.×9-in., two cylinder B. D. machine. 2 D. P. ammonia condensers, 10 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel. Gas and liquid headers. 1 Ammonia oil separator, 12 in.×36 in. 1 Ammonia receiver, 20 in.×7 ft. Ammonia connections: 30 ft., 2 in. E. H. steel pipe. 5 2 in. ammonia ells, C. F. B. & G. 2 2½-in. ammonia ells, C. F. B. & G. 2 Pr. 2-in. ammonia flanges, B. & G 10 ft., ¾-in. E. H. pipe. 1 ¾-in. ammonia screw tee. 1 Pr. ¾-in. flanges, B. & G. Gage connections. B. D. machine with extended base plate and outboard bearing. Machine with extended base plate arranged for direct vertical engine drive. Machine without entended base plate arranged for direct vertical engine drive. Machine without extended base plate arranged for horizontal uniflow engine drive. Machine without extended base plate or flywheel but with shaft length suitable for engine type synchronous motor drive. No swell in shaft permissible. Outboard bearing included. Motor rotor must have sufficient weight to furnish necessary flywheel effect. Motor rotor must be supplied with split hub or split rotor.

No exciter pulley included in above.

NOTE.—The gage connections furnished with 10 in.×10 in. and 12 in.×12 in. Enclosed Type Machine H. S. consist of the following:

40 ft. ¼-in. E. H. steel pipe, 2 ¼-in. ammonia valves screwed.
2 ¼-in. ammonia ells, screwed. 4 Prs. ¼-in. ammonia flanges, B. & G.

TABLE 173

Sand Filters, Weights

Tons of Ice	Number of Filters	Size of Filters, Inches	Style	Weight, Pounds
1	1	12 × 50	C	700
2	1	12 × 50	C	700
3	1	12 × 50	C	700
4	1	12 × 50	C	700
5	1	12 × 50	C	700
6	1	12 × 50	C	700
8	1	16 × 51	C	1,000
10	1	16 × 51	C	1,000
12	1	18 × 52	C	1,350
15	1	18 × 52	C	1,350
20	1	24 × 53	C	2,200
25	1	24 × 53	C	2,200
30	1	24 × 53	C	2,200
35	1	24 × 53	C	2,200
40	2	24 × 53	C	4,400
50	2	24 × 53	C	4,400
60	2	30 × 61	I	5,700
80	2	30 × 61	I	5,700
100	2	36 × 62	I	8,100
120	3	30 × 61	I	8,550
150	3	36 × 62	I	12,150
200	4	36 × 62	I	16,200

TABLE 174

Ice Freezing Cans, Specifications, Weights

Number of Cans per Ton per 24 Hr., 19 Lb. Suction, 250 Ft. 1¼-in. Coils per Ton		Weight of Ice Cake, Pounds	Size of Cans, Inches				Space Required in Tank for Each Can, Inches			Weight of Can, Pounds
Expansion freezing system	Flooded freezing system		Width at top	Length at top	Height over all	Gage of galvanized iron	Wide	Long	High	
22	20¼	25	4	9	24	18	7	11	27	20
S-17½	15¾	50	5	12	32	16	8	14	35	27
25	23	50	6	10	32	16	9	12	35	26
22	20¼	50	8	8	32	16	11	10	35	26
14½	13	60	5	14	32	16	8	16	35	29
S-22	20¼	100	8	16	32	16	11	18	35	42
S-23	21	200	11½	22½	32	16	14½	24½	35	60
18½	16½	200	14½	14½	36	16	17½	16½	39	55
S-15½	14	300	11½	22½	45	16	14½	24½	48	70
S-11½	10½	400	11½	22½	58	14	14½	24½	61	105

Note.—Cans marked "S" are Association Standard.

CHAPTER XXII

ELECTRIC MOTORS

For some time it has been considered essential that the refrigerating compressor be capable of having a widely varying rotative speed, but this is seldom demanded at the present time. When a variation of capacity is required it can be obtained by varying the number of

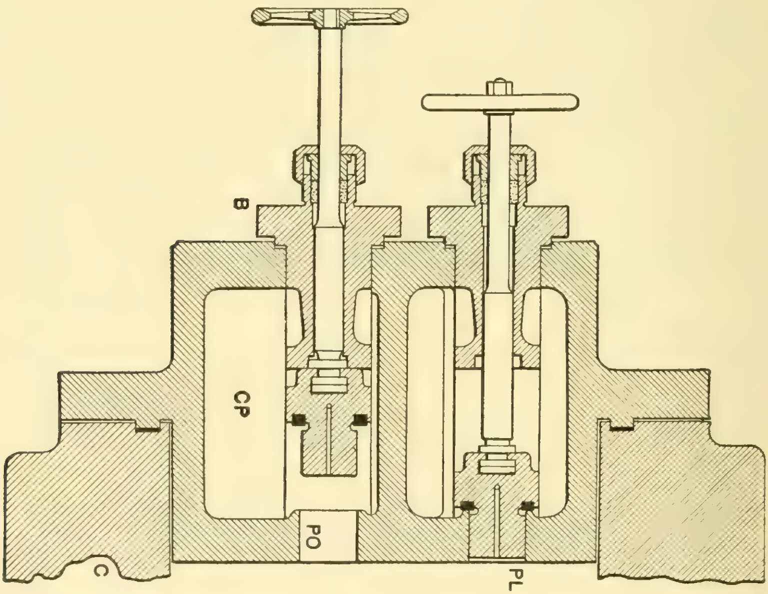


Fig. 385.—The Clearance Pocket.

machines applied to the load, varying the number of cylinders in action by disconnecting one cylinder of a duplex compressor, by increasing or decreasing the amount of clearance in the cylinder by the expedient of clearance pockets (Fig. 385), and in the smaller sizes by making the compressor automatic. In Chapter II it was shown that the amount of clearance affected very decidedly the capacity of the compressor, but

that the economy of operation, i.e., the horsepower per ton, was not appreciably affected by clearances from 4 to 8 per cent or more. Rotary speed control, therefore, is not of primary importance whereas the requirements of a good electric motor are a good starting torque and high efficiency during full speed operation. Before going into the details of the various types of motors a few electrical principles will be considered.

Power.—True power in an alternating electric circuit is the average value of the products of the coincident instantaneous values of the current and the voltage for a complete circuit. Power factor is the ratio of the true power, KW, to the apparent power, kva. This is expressed as

$$PF = \frac{KW}{kva}$$

The power in an electric circuit is:

$$\begin{aligned} \text{Watts} &= EI \times PF \text{ for single phase} \\ &= 2EI \times PF \text{ for two phase} \\ &= 1.732\, EI \times PF \text{ for three phase,} \end{aligned}$$

where I = the effective line current and E = the effective volts between the terminals. The effect of changes of voltage affects the starting torque as the *square* of the voltage applied to the primary.

Three-phase circuits may be connected in delta (Δ) or in star (Y). If three transformers are connected in delta, the voltage across each transformer is the same as the line voltage, whereas the current in each line wire is $\sqrt{3}$ times the current in the transformer winding. If three transformers are connected in star the current will be the same as the line current, but the voltage will be $\sqrt{3}$ times the voltage across the transformer windings.

Wire and Wiring.—The size of conductors can be obtained by the formula, for single and two phase with unit power factor at the motor:

$$cm = \frac{21.6DI}{e},$$

where

cm is the size of the copper wire in circular mils;
D is the length of the circuit, in feet;
I is the current, in amperes;
e is the voltage drop in the length of circuit D.

For three phase use the constant 10.8 where e is the voltage drop per wire. The line voltage at the motor divided by $\sqrt{3}$ is the supply voltage minus the voltage drop e.

TABLE 175

DIMENSIONS, WEIGHTS AND RESISTANCES OF PURE COPPER WIRE (BARE)—BROWN AND SHARPE GAGE

Gage Number	Diameter, Inch	Area		Weight, Pounds per 1000 Ft.	Length, Feet per Pound	Resistance at 75 Deg. F.			Carrying Capacity, Amperes	
		Circular mils (d^2) mils = .001 inches	Square mils (d^2) × .7854			Length, Feet per ohm	R ohms per 1000 ft.	Ohms per pound	Rubber insulation	Other insulation
	1.152	1,000,000	785,400	3050.0	.3275	95,100.0	.01051	.000003442	650	1000
	1.035	800,000.0	628,000.0	2440.0	.4100	76,100.0	.01313	.000005380	550	840
	.963	700,000.0	549,500.0	2135.0	.4680	66,600.0	.01501	.000007030	500	760
	.891	600,000.0	471,000.0	1830.0	.5460	57,100.0	.01751	.000009579	450	680
	.819	500,000.0	392,000.0	1525.0	.6550	47,500.0	.02101	.00001376	390	590
	.728	400,000.0	313,900.0	1220.0	.8100	38,050.0	.02627	.00002155	330	500
	.590	250,000.0	196,000.0	762.0	1.32	23,750.0	.04203	.00005600	275	412
0000	.4600	211,600.0	166,190.0	639.3	1.56	20,383.0	.04906	.00007673	225	325
000	.4096	167,805.0	131,790.0	507.0	1.97	16,165.0	.06186	.00012039	175	275
00	.3648	133,079.0	104,520.0	402.0	2.49	12,820.0	.07801	.00009423	150	225
0	.3248	105,538.0	82,887.0	318.8	3.14	10,166.0	.09838	.00038500	125	200
1	.2893	83,694.0	65,733.0	252.8	3.99	8,062.3	.12404	.00048994	100	150
2	.2576	66,373.0	52,130.0	200.5	4.99	6,393.7	.15640	.00078045	90	125
3	.2294	52,634.0	41,339.0	159.0	6.29	5,070.2	.19723	.0012406	90	100
4	.2043	41,742.0	32,784.0	126.1	7.93	4,021.0	.24869	.0019721	70	90
5	.1819	33,102.0	25,998.0	100.0	10.00	3,188.7	.31361	.0031361	55	80
6	.1620	26,250.0	20,617.0	79.32	12.61	2,528.7	.39546	.0049868	50	70
7	.1442	20,816.0	16,349.0	61.90	15.90	2,005.2	.49871	.0079294	38	54

8	.1284	16,509.0	12,960.0	49.88	20.05	1,590.3	.62881	.0102608	35	50
9	.114	13,094.0	10,284.0	39.56	25.28	1,261.3	.79281	.020042	28	38
10	.1018	10,381.0	8,153.0	31.37	31.38	1,000.0	1.0	.031380	25	30
11	.0907	8,234.0	6,467.0	24.88	40.20	1,000.0	1.2607	.050682	20	27
12	.0808	6,529.0	5,128.0	19.73	50.69	629.02	1.5898	.080585	20	25
13	.0719	5,179.0	4,067.0	15.65	63.91	298.83	2.0047	.12841	14	20
14	.0640	4,106.0	3,225.0	12.44	80.38	395.60	2.5278	.20322	15	20
15	.0570	3,256.0	2,557.0	9.84	101.63	321.02	3.1150	.31658		
16	.0508	2,582.0	2,028.0	7.81	128.14	248.81	4.0191	.51501	6	10
17	.0452	2,049.0	1,608.0	6.19	161.59	197.30	5.0683	.81900		
18	.0403	1,624.0	1,275.0	4.91	203.76	156.47	6.3911	1.3023	3	5
19	.0358	1,287.0	1,011.0	3.88	257.47	123.99	8.0654	2.0759		
20	.0319	1,021.0	802.2	3.09	324.00	98.40	10.163	3.2926		
21	.0284	801.1	636.2	2.45	408.56	78.06	12.815	5.2355		
22	.0253	642.7	504.7	1.94	515.15	61.91	16.152	8.3208		
23	.0225	509.4	500.1	1.51	649.66	49.08	20.377	13.238		
24	.0201	404.0	317.3	1.22	819.21	38.01	25.695	21.050	In lighting work no smaller than No. 14 is used, except for fixtures.	
25	.0179	320.4	251.6	.97	1,032.9	30.86	32.400	33.466		
26	.0159	254.0	199.5	.77	1,302.6	24.46	40.868	55.235		
30	.0100	100.5	78.9	.30	3,293.9	9.681	103.30	340.25		
36	.0050	25.0	19.6	.08	13,238.0	2.408	415.24	5,497.4		
40	.0031	9.9	7.7	.03	33,157.0	.952	1049.7	34,823.0		

For sizes above No. 0000 the table refers to standard copper cable. Due to the conditions imposed by proper "laying" of the strands of figure for circular mileage, diameter, resistance, etc., can be given with only approximate accuracy.

One foot of pure copper wire of one circular mil cross-sectional area (diameter .001-in.) has a resistance of 10.74 ohms at 15 deg. C. or 59 deg. F.

The copper wire table (Table 175) has simple relations for approximate accuracy. A copper wire three sizes larger (B. and S. gage) than another wire has half the resistance, twice the weight and twice the area. A wire which is ten sizes larger than another has one-tenth the resistance, ten times the weight and ten times the area. The larger sized wire is specified in circular mils, and the number of feet per ohm is one-tenth of the circular mileage. To find the weight in pounds, drop four ciphers from the number of circular mils, and multiply by the weight of the No. 10 wire.

The Direct Current Motor.—The direct current motors are classified according to the field windings as shunt, compound and series motors. In the shunt wound motors the field is excited by connecting its winding with a field rheostat in series with the supply line. As the load increases the demand for current increases and a slight decrease in speed results in shunt motors of good design.

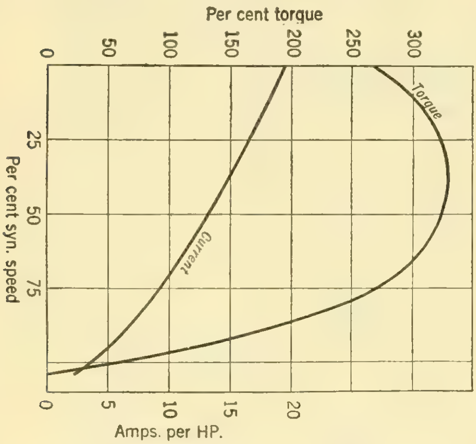


Fig. 386.—Repulsion Motors.

The *series wound* motor has its field winding in series with the armature winding, and all of the current passes through the field coils. The motor speed will vary inversely as some power of the motor load, and these motors are always direct connected by coupling, chain or gear to the load, and even then they are always under the control of the operator. They are capable of starting and accelerating heavy loads with lower current consumption than any other type of motor, and they are especially adapted for use on cranes, hoists, etc.

The cumulative *compound* wound motor is, as the name implies, a combination of both shunt and series windings for the field excitation. These motors are not constant speed, but they are used where the required torque varies considerably, being high at starting but where at the same time the speed limiting characteristics of the shunt motor are desirable.

The Alternating Current Motors.—In general, the alternating current motors are classified as induction, repulsion-induction and synchronous. The alternating current induction motors are classified accord-

ing to the secondary windings as squirrel cage or wound rotor. At the present times there is no satisfactory adjustable speed alternating current motor in the sense of the direct current motor, and no alternating current motor should be placed on loads requiring a variable speed.

The drop of speed in the induction motor from no load to full load is called the "slip," and this is proportional to the amount of resistance in the motor windings (Table 176). If a motor has a small slip it is considered a constant speed motor, and it compares with the direct current shunt wound motor in its speed torque characteristics. If considerable resistance is placed in the armature the slip becomes comparatively high and the speed torque characteristics are comparable with those of the compound or even with the series wound direct current motors, thereby securing a motor more adaptable for varying speed work.

Low speed motors, having a slip of from 2 to 5 per cent, of either the so-called squirrel or the wound rotor type are adapted to constant speed work, and the selection of the squirrel cage or the wound rotor motor depends on the requirements as regards the starting torque. The low slip squirrel cage motor usually has starting torques not less than the following, with full line voltage applied:

2 and 4 pole motors have 150 per cent of full load torque;
6 pole motors have 135 per cent of full load torque;
8 pole motors have 125 per cent of full load torque;
10 pole motors have 120 per cent of full load torque;
12 pole motors have 115 per cent of full load torque;
14 pole motors have 110 per cent of full load torque.

It requires from 3 to $4\frac{1}{2}$ times the full load current from the line to develop full load torque with the reduced voltage applied at starting, in the case of the squirrel cage, and only $1\frac{1}{4}$ times the full load current with the wound rotor when the proper resistance is inserted between the collector rings. The maximum running torque, or the so-called pull out torque, for all polyphase induction motors with rated impressed voltage is not less than 200 per cent of full load torque.

Squirrel cage motors of somewhat higher slip, from 8 to 12 per cent, are used where the starting torque required is large compared with the running torque, or where flywheels are employed. By varying the external resistance on a wound rotor motor a condition can be created which will practically duplicate the speed torque characteristics of the various classes of squirrel cage motors, but the current is approximately the same for all speeds.

The single phase *repulsion induction* constant speed motor is good for small capacities where a high starting torque is required as well as the lowest possible starting current. Such motors give starting torques of from 2 to 4 times the full load torque, and the maximum running torque is from $1\frac{3}{4}$ to 2 times or more the full load torque.

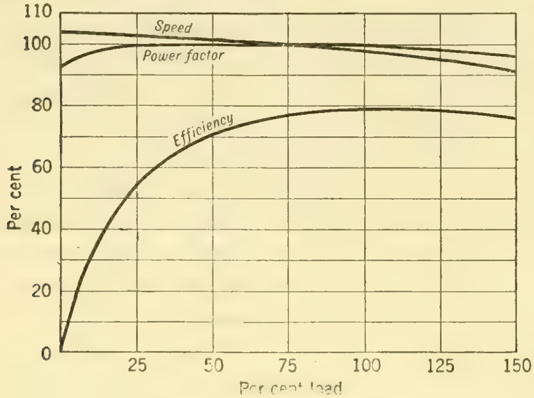


FIG. 387.—Repulsion Motors.

Figure 386 shows the operating characteristics of this type of motor, and Fig. 387 gives the efficiency and the power factor. The power factor may be made leading at light loads but at a loss of efficiency so that leading power factor is seldom called for.

Direct Connected Electric Motors.—In contrast to the ammonia compressor the direct connected carbonic compressor is a problem by itself due to the unbalance occasioned by a difference of about 25 per cent in the net area of the head and crank ends and the very high suction pressures of 300 to 500 lb. per sq. in. According to McLenegran[1] a double-acting, single-cylinder carbonic compressor will require

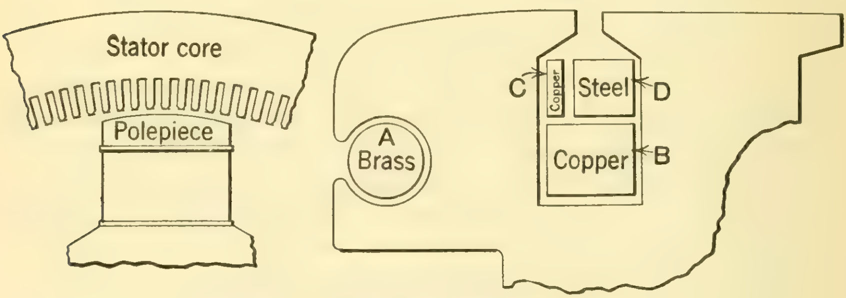


FIG. 388.—Pole Faces of Synchronous Motors.

$37\frac{1}{2}$ per cent of full load torque simply to overcome the unbalance during the starting period. After full speed is attained the flywheel will partly smooth out the unbalance.

[1] D. W. McLenegran, Refrigerating Engineering, January, 1927.

If the double-acting compressor is one with two cylinders with cranks at 90 degrees, the unbalance will be reduced to about 21 per cent, whereas the duplex double-acting compressor reduces the unbalance to zero provided the cranks are at 180 degrees.

Flywheel Requirements.—Considerable interest has been shown on the subject of flywheel requirements for electric driven refrigerating machines.[2] The problem appears to be one of reducing the current pulsations to a minimum, to get the required flywheel effect in the rotor of the motor if possible and to have a flywheel, or its equivalent, which will come into step easily on excitation. Stevenson has worked out a number of curves for particular designs of compressors which reduce the labor in the calculation of the required size of the flywheel.

The Synchronous Motor.—A number of factors have been responsible for the popularity of the synchronous motor. It lends itself to direct connection having a comparatively large air gap with respect to the induction motor (which has from 0.02 in. to 0.05 in.) and so it can take an appreciable wear of the bearings without danger of rubbing of the stator and rotor. The synchronous motor can give unity power factor or even a leading power factor without any more difficulty than an over excitation of the field. Depending on the type of compressor, the necessary flywheel effect for satisfactory operation may be incorporated in the rotor of the slow speed synchronous motor, thereby saving in space and the cost of material. It is both feasible and desirable also to throw the motor direct on the line, thus eliminating auto transformers and tandem switches, for small capacities only.

The synchronous motor, from its name, means no slip whatsoever. The torque is produced by a slight lag, some $3\frac{1}{2}$ to 6 electrical degrees, which enables it to carry the load in a manner similar to the action of torsion in a steel shaft. Such motors to be self-starting must come up to the neighborhood of synchronous speed by means of an induction motor feature in the design. This consists of brass or copper bars, or iron or steel conductors inside brass tubes, placed in slots in the pole faces, the bars being connected together by means of end rings. During the starting period the direct current excitation is removed, and the revolving flux produced by the alternating current in the armature windings reacts on the squirrel cage windings as in the induction motor. As the induction motor cannot reach synchronous speed the final

[2] Theodore Schou, The Present Status of Synchronous Motors for Direct Connection to Compressors, Amer. Society Refrigerating Engineers, May 26, 1922.

A. R. Stevenson, The Flywheel Problem in Compressors Direct Connected to Synchronous Motors, Refrigerating Engineering, October, 1924.

"pull in" is obtained when the direct current excitation is applied, which it will do if the speed is within 2 to 5 per cent of synchronism, depending on the kind of load and the amount of flywheel effect which it is necessary to accelerate. In the case of reciprocating compressors, which give a varying resisting torque, there is a periodic variation in the displacement angle of the rotor, and a corresponding fluctuation in the current drawn from the supply. To make this variation a minimum,

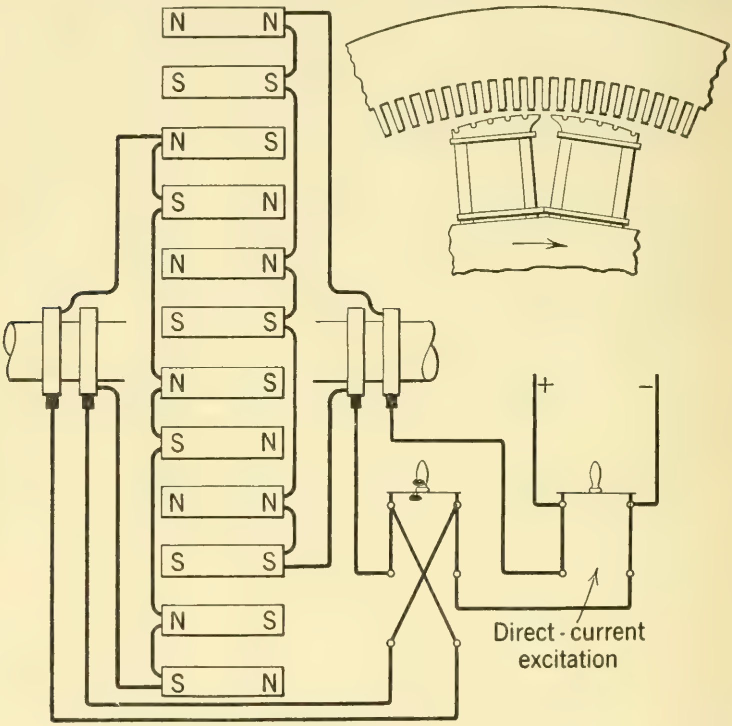


Fig. 389.—Two Speed Synchronous Motor.

the weight of the flywheel should be sufficient, but too great a weight will prevent a proper starting of the motor.

Figure 388 shows the arrangement of one type of pole-face windings. About 35 per cent of the full load torque is required to start compressors unloaded. Torques of 40 per cent will bring the average compressor up to 95 per cent synchronous speed in less than 30 seconds. The modern type of motor will deliver not less than 140 per cent of the full load torque before falling out of synchronism. The modern tendency to decrease the air gap decreases the required amount of direct current excitation.

Figure 389 shows a design of a two-speed synchronous motor. At times such a variation of speed is desirable, and the efficiency of such a motor operating at speeds of, say, 600 and 300 r.p.m. in the medium sizes, is about 95 per cent at both speeds.

TABLE 176

Speed of Rotary Field for Different Numbers of Poles and for Various Frequencies

Number of Poles	Speed of Revolving Magnetism, in R.p.m., When Frequency Is					
	25	30	$33\frac{1}{3}$	40	50	60
2	1500	1870	2000	2400	3000	3600
4	750	900	1000	1200	1500	1800
6	500	600	667	800	1000	1200
8	375	450	500	600	750	900
10	302	360	400	480	600	720
12	250	300	333	400	500	600
14	214	257	286	343	428	514
16	188	225	250	300	375	450
18	167	200	222	267	333	400
20	150	180	200	240	300	360
22	136	164	182	217	273	327
24	125	150	167	200	250	300

Slip of Induction Motors

Capacity of Motor Horse Power	Slip, at Full Load, Per Cent		Capacity of Motor Horse Power	Slip, at Full Load, Per Cent	
	Usual limits	Average		Usual limits	Average
$\frac{1}{8}$	20–40	30	15	5–11	8
$\frac{1}{4}$	10–30	20	20	4–10	7
$\frac{1}{2}$	10–20	15	30	3–9	6
1	8–20	14	50	2–8	5
2	8–18	13	75	1–7	4
3	8–16	12	100	1–6	$3\frac{1}{2}$
5	7–15	11	150	1–5	3
$7\frac{1}{2}$	6–14	10	200	1–4	$2\frac{1}{2}$
10	6–12	9	300	1–3	2

TABLE 177

Wire Capacity of Conduits

Size of Wire B. and S. Gage	Diameter Rubber Insul. Double Braid in 32nds. In.	Number of Wires in One Conduit Size of Unlined, Wrought-iron, Conduit for 0–600 Volts, Rubber Insulated Double-braid Wires								
		1	2	3	4	5	6	7	8	9
14	8	½	½	¾	¾	1	1	1	1	1¼
12	9	½	¾	¾	¾	1	1	1¼	1¼	1¼
10	10	½	¾	1	1	1	1¼	1¼	1¼	1¼
8	11	½	1	1¼	1¼	1¼	1¼	1¼	1½	1½
6	14	½	1	1¼	1¼	1½	1½	1½	2	2
3	15	¾	1¼	1¼	1¼	1½	2	2	2	2½
4	16	¾	1¼	1¼	1½	1½	2	2	2	2½
3	17	¾	1¼	1¼	1½	2	2	2	2½	2½
2	19	¾	1½	1½	2	2	2½	2½	2½	3
1	21	1	1½	2	2	2½	2½	3	3	3
0	23	1	2	2	2	2½	3	3	3	3½
00	24	1	2	2	2½	2½	3	3	3	3½
000	26	1¼	2	2½	2½	3	3	3½	3½	3½
0000	28	1¼	2	2½	3	3	3½	3½	3½	
200,000 C. M.	29	1¼	2	2½	2½					
250,000 "	30	1¼	2½	2½	3					
300,000 "	32	1½	2½	3	3					
350,000 "	1–2	1½	3	3	3½					
400,000 "	1–3	1½	3	3	3½					
450,000 "	1–5	1½	3	3	3½					
500,000 "	1–6	1½	3	3	3½					

Two-phase vs. Three-phase.—Due to the superior distribution of the stator windings, three-phase motors are slightly more efficient at all loads than two-phase motors of corresponding size. The power-factor is also higher especially at light loads, and the starting torque with full-load current is greater. Furthermore, for equal conditions of load and voltage, the amount of copper required in the distributing system is less. Consequently, whenever service conditions will permit, three-phase motors are preferable to two-phase.

TABLE 178

WIRE SIZES FOR ALTERNATING CURRENT MOTORS

Sizes of wire, fuses and switches for use with squirrel cage continuous rated induction motors. Voltage drop not considered. Use next larger wire for motors below 600 r.p.m. Use last column for wound rotor motors.

Horse Power	Average Ampere Rating	Wire Size Code	Switch Amperes	Starting Fuse	Running Fuse	Wire Size Based on 125 Per Cent Load
220-volt Motors, 3-phase, 60 Cycles						
1	3.5	14	15	10	5	14
3	9.5	12	25	25	15	14
5.0	15.4	10	50	35	20	12
7.5	22.4	8	50	45	30	8
10.0	29.0	6	75	65	40	7
15.0	42.5	4	100	100	60	5
20.0	55.0	2	150	130	75	4
25.0	68.0	1	150	150	95	2
30.0	80.0	0	200	200	100	1
40.0	105.0	00	300	225	150	00
50.0	130.0	000	300	250	175	000
		Mcm.				Mcm.
75.0	192.0	300	400	400	275	250
100.0	252.0	450	500	550	350	400
150.0	368.0	800	800	850	550	600
200.0	484.0	1000	1000	1150	725	900
440-volt Motors, 3-phase, 60 Cycles						
1.0	1.8	14	15	5	3	14
3.0	41.8	14	15	15	7	14
5.0	7.8	14	25	20	10	14
7.5	11.0	10	25	25	15	14
10.0	15.0	9	50	35	20	12
15.0	21.0	8	50	50	30	10
20.0	27.0	7	75	75	40	8
25.0	34.0	6	75	75	50	7
30.0	40.0	5	75	90	55	6
40.0	52.0	3	150	135	75	4
50.0	65.0	2	150	175	90	3
75.0	96.0	0	300	225	140	0
100.0	126.0	000	300	275	175	00
		Mcm.				
150.0	184.0	250	400	400	275	0000
						Mcm.
200.0	242.0	400	500	550	350	400

CHAPTER XXIII

STEAM AND OIL ENGINES

Previous to 1915, generally speaking, and 1910 in particular, the steam engine was almost the only source of power in ice making plants. At the present time (1927) steam is seldom used except in special cases, such as the packing plants, some isolated ice-making plants, especially if the water available at the plant requires distillation in order to make it fit for can ice making, and in the majority of the larger hotels and apartment houses.

Whereas in the earlier days, when coal was comparatively cheap, the Corliss engine was used almost exclusively, now it is usual to use the more economical forms of prime movers. The exceptions would be in cases such as hotel refrigeration where a heat balance is attempted and the exhaust steam is used for heating purposes in the winter and an electric motor drive is used at times when steam is not required for heating.

When economy of steam is required, and a relatively small engine is to be used, the poppet valve and the uniflow steam engines are the usual types which are likely to be selected.

The Uniflow Steam Engine.—One of the great losses in economy in the steam engine is that due to cylinder condensation and re-evaporation. These losses are occasioned by the absorption of heat by the cylinder walls, ports and the piston head. When saturated steam is used the exposed metal causes a partial condensation of the steam up to the point of cut-off—the walls and ports being heated accordingly—and then during the expansion of the steam after the point of cut-off these same metals begin to heat and re-evaporate the steam as soon as the steam temperature falls below that of the cylinder walls. As heat is taken from the cylinder walls and head its temperature falls to a point that is relatively cold as compared with the incoming steam at the beginning of the next stroke. The net result of this heat exchange between the steam and the metals of the engine is the use of much more steam than is required for the work performed. In order to overcome these losses it is possible to use superheated steam or a design, as in the uniflow engine, which does not permit the cooling of the engine to

develop to the extent that it does in the slide valve or the usual four-valve engines. In the uniflow design the steam is exhausted through ports in the cylinder barrel which are uncovered by the piston at the end of its stroke. The piston has to be especially long—slightly less than the length of the stroke. The uniflow engine is usually designed to operate

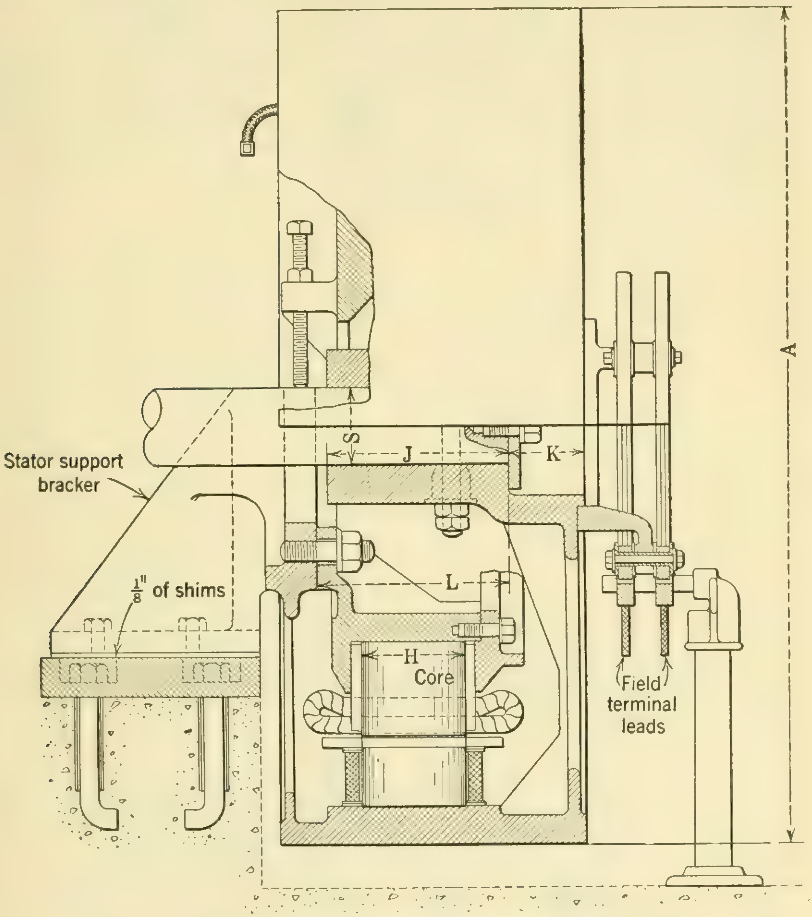


Fig. 390.—Flywheel Type Synchronous Motor.

both condensing and non-condensing. and by the use of a special device the engine may operate either manner automatically.

The Water Rate.—The performance curves (Figs. 392 and 395) show that with saturated steam the water rate is less than 14 lb. per i.hp.hr., but this steam consumption rate is much improved with the use of

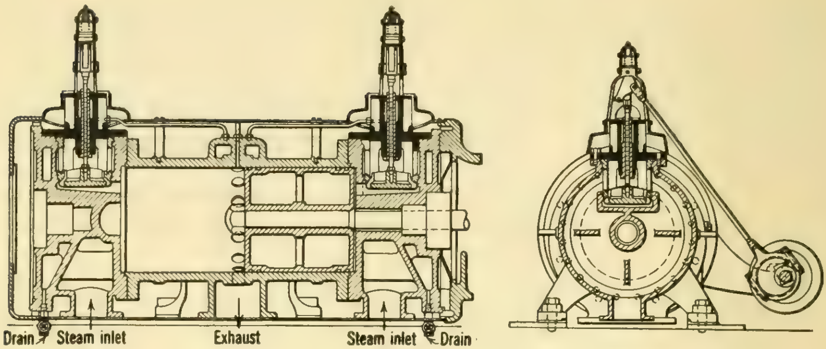


FIG. 391.—Uniflow Steam Engine.

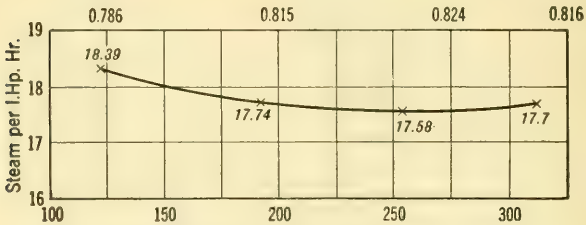


FIG. 392.—Water Rate Uniflow Engine.

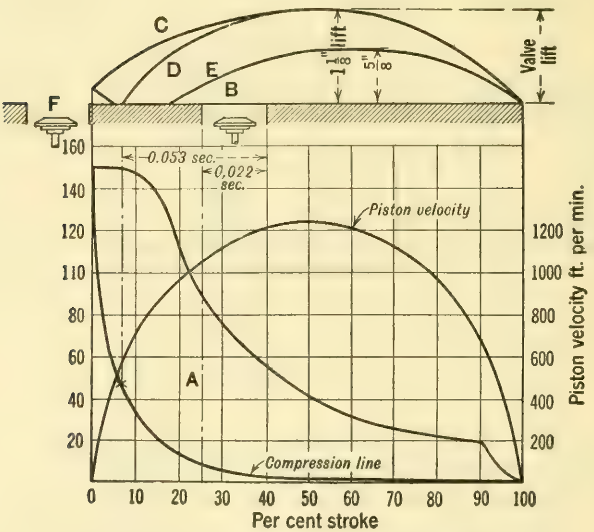


FIG. 393.—Piston Velocity of Uniflow Engine.

superheated steam. At full load and with 100 deg. F. superheat the water rate is less than 10.5 lb., and with 200 deg. superheat it is less than 9 lb. The action of the superheat is to tend to prevent the condensation of the steam, and the rate of absorption of heat by the materials of the engine is reduced very greatly on account of the lower value of the coefficient of heat transfer from metals to a gas than to a wet vapor. Sizes of uniflow steam engines are given in Tables 180 and 183.

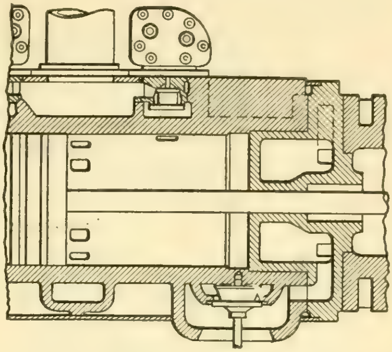


Fig. 394.—Double Seated Exhaust Uniflow Steam Engine.

The Oil Engine.—One of the recent outstanding developments in prime movers applied to the refrigerating industry has been that in regard to the oil engine of the Diesel and the semi-Diesel types. The economy of the Diesel engine is remarkable, being as low as 0.4 lb. of fuel oil per b.hp.hr. This type of engine has become reliable, and the maintenance cost is claimed to be less than that of the majority of small steam plants. One factor, however, is that of the operator who is not as numerous as the steam engineer, and this might cause considerable annoyance in isolated plants, except in the case of the simpler designs of semi-Diesel, hot bulb, etc., types, which are becoming quite reliable.

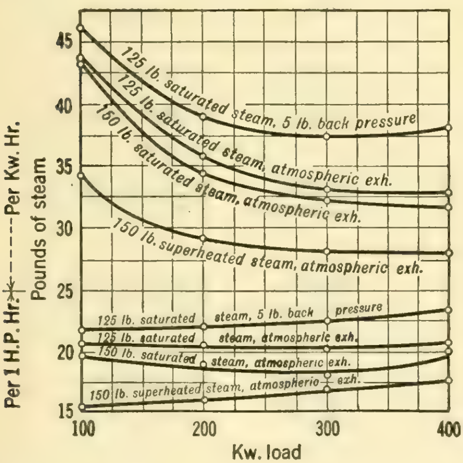


Fig. 395.—Steam Consumption Vertical Uniflow Steam Engine.

Generally speaking, it can be said that the oil engine usually employed for refrigerating plants is direct connected, although the belt drive is still found. The oil engine may be horizontal or vertical, four-cycle or two-cycle, air injection or airless injection of the fuel, full or semi-Diesel, all combined with various methods of cooling, and with details of the

TABLE 179

Tabulation of Approximate Dimensions—Sizes, Ratings and Steam Consumption at 125 Lb. Steam

Center Crank Flat Guide, Kw.	R.p.m.	Cylinder, Inches	Total Weight, Pounds	Shaft Weight, Pounds	Length		Width		Cubic Feet in Foundation	Steam Rate per 1 Hp. Hr. at 125 Lb. Saturated Steam, Atmospheric Exhaust—Proportional Load in Kw.				
					Ft.	In.	Ft.	In.		1/4	1/2	3/4	4/4	5/4
150	277	12×13	15,500	1025	12	1	8	7	350	21.5	20.1	19.8	20.3	21.8
60	277	12×13	15,500	1025	12	1	8	7	350	21.0	19.9	19.9	21.2	23.6
75	257	14×15	22,000	2100	13	9	9	9	590	21.2	19.8	19.5	20.1	21.7
100	257	15×16	23,500	2100	14	2	10	7	710	20.7	19.5	19.4	20.4	22.3
125	257	16×16	24,000	2100	14	2	10	7	710	20.5	19.5	19.6	21.1	23.6
150	225	17×18	28,500	2500	15	6	11	1	960	20.2	19.3	19.7	21.7	24.4
175	225	19×20	37,000	3350	17	0	12	5	1130	20.7	19.3	19.4	20.2	22.1
For 3 lb. B. P.	add......	....	....	....	....	....	....	....	....	1.4	1.3	1.2	1.1	1.1
Side Crank Flat Guide, Kw.														
50	277	12×13	13,500	1200	12	0	9	8	430	21.5	20.1	19.8	20.3	21.8
60	277	12×13	13,500	1200	12	0	9	8	430	21.0	19.9	19.9	21.2	23.6
75	257	14×15	18,000	1700	13	4	11	0	580	21.2	19.8	19.5	20.1	21.7
100	257	15×16	21,000	2800	13	9	11	0	740	20.7	19.5	19.4	20.4	22.3
125	257	16×16	22,000	2800	13	9	11	0	740	20.5	19.5	19.6	21.1	23.6
Side Crank Bored Guide, Kw.														
100	257	16×16	22,500	2800	13	9	11	0	740	21.3	19.8	19.4	19.8	21.2
125	257	17×16	22,500	2800	13	9	11	0	740	20.8	19.5	19.4	20.2	22.1
125	225	17×18	31,000	3700	16	5	12	6	1300	20.7	19.4	19.3	20.1	22.0
150	225	17×20	32,000	3700	17	0	12	6	1300	20.4	19.3	19.3	20.7	23.0
175	225	19×20	36,000	3700	17	0	12	6	1300	20.5	19.3	19.3	20.2	22.2
175	225	19×20	38,200	4300	17	0	13	1	1340	20.5	19.3	19.3	20.2	22.2
175	200	21×20	38,700	4300	17	0	13	1	1340	20.9	19.4	19.2	19.8	21.6
175	200	19×21	39,700	4300	20	4	13	1	1340	20.3	19.2	19.3	20.9	23.2
200	200	21×21	40,200	4300	20	4	13	1	1340	20.5	19.3	19.3	20.2	22.1
200	180	21×21	42,500	4300	20	4	13	1	1340	20.3	19.3	19.6	21.0	23.4

TABLE 180

Specifications of Anderson Diesel Engines (Type KD)—Heavy Duty

R.p.m. = 257. Piston Speed = 771 Ft. per Minute.
6-cylinder Engines Have One Flywheel

Horse Power Rating	Kilo-watt Capacity	Number of Cylinders	Flywheel Diameter, Width, Inches	Weight, Pounds	Exhaust Opening, Inches	Piston Length, Inches	Weight, Pounds	Floor Space, Inches	Height above Floor		Concrete Cubic Yards of
									Ft.	In.	
50	33	1	62×10	3300	8	24	19,100	62× 81	9	11	7.9
60	40	1	66×10	3600	8	28	21,400	66× 87	10	1½	8.7
100	67	2	62×10	3300	8	24	25,700	62×109	9	11	11.3
120	80	2	66×10	3600	8	28	32,600	66×120	10	1½	12.5
150	100	3	62×10	3300	8	24	35,100	62×137	9	11	14.7
180	120	3	66×10	3600	8	28	42,400	66×153	10	1½	16.2
200	134	4	62×10	3300	8	24	39,500	62×166	9	11	18.1
240	160	4	66×10	3600	8	28	51,200	66×186	10	1½	20.0
300	200	6	62×10	3300	8	24	59,000	62×222	9	11	24.8
360	240	6	66×10	3600	8	28	64,000	66×252	10	1½	27.0

Note.—The shipping weights above are intended to cover shipment where engines are sent out with the average equipment.

TABLE 181

Fairbanks-Morse Diesel Engines

Horse Power	Cylinders	R.p.m.	Floor Space: Standard and belted electric				Floor Space: Direct-connected electric engine and generator				Height above Floor	
			Ft.		Ft.	In.	Ft.		Ft.	In.	Ft.	In.
40	1	300	8	×	9	7	8	×	13	7½	8	7
60	1	257	8	×	11	3	8	×	13	6	9	6
80	2	300	8	×	12	0	8	×	15	3½	8	7
120	2	257	8	×	14	7	8	×	16	8½	9	6
180	3	257	8	×	17	10	8	×	19	10	9	6
240	4	257	8	×	21	0	8	×	22	7	9	6
360	6	257	8	×	28	0	8	×	30	6¾	9	6

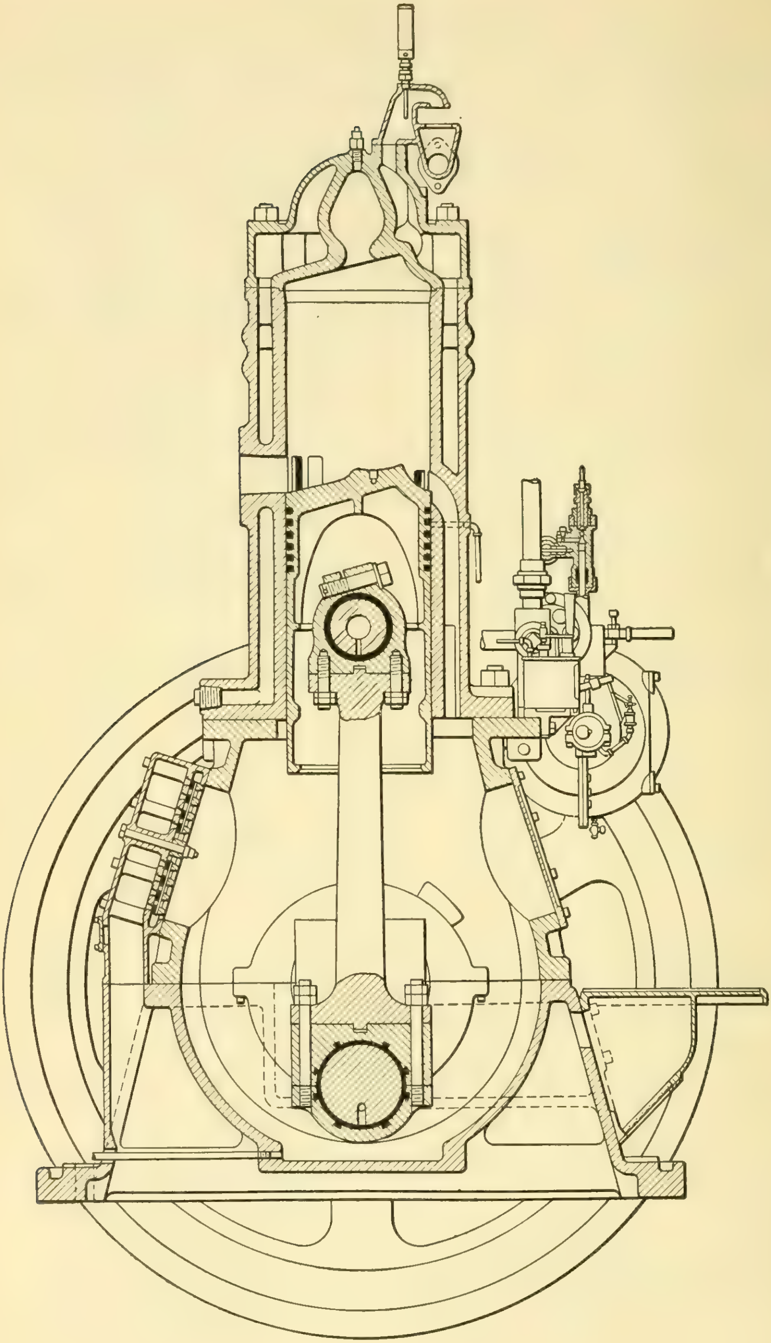


FIG. 396.—Anderson Oil Engine.

cylinder head, fuel nozzle, governor, piston head and details governing the method of ignition.

Figure 398 gives a typical fuel consumption curve of a 100-hp. oil

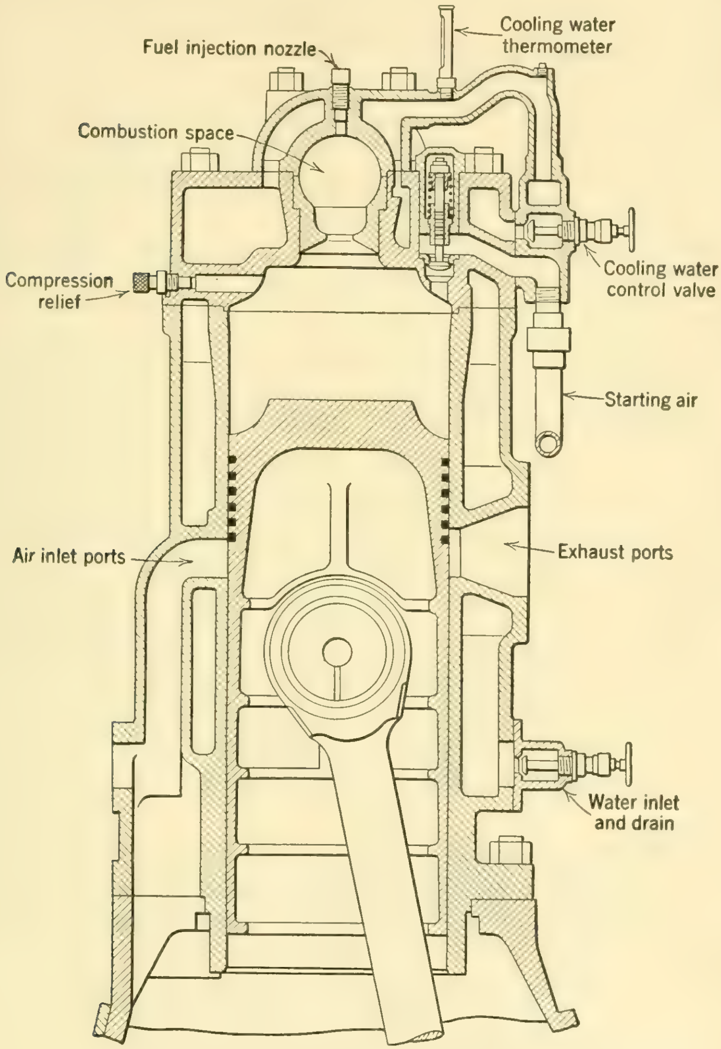


Fig. 397.—Fairbanks-Morse Oil Engine.

engine. The oil engine-driven plant, consisting of two 100-hp. oil engines, would cost with auxiliaries, including a 60-kw. generator unit, exhaust silencers, starting unit, oil storage tank, foundations, etc., about $17,500

more than the electrically driven plant consisting of two 100-hp. synchronous motors with exciters and control panels. This cost, figured on a basis of 15 per cent overhead, to include interest, depreciation and taxes would amount to $2625 per year. To make the oil engine economi-

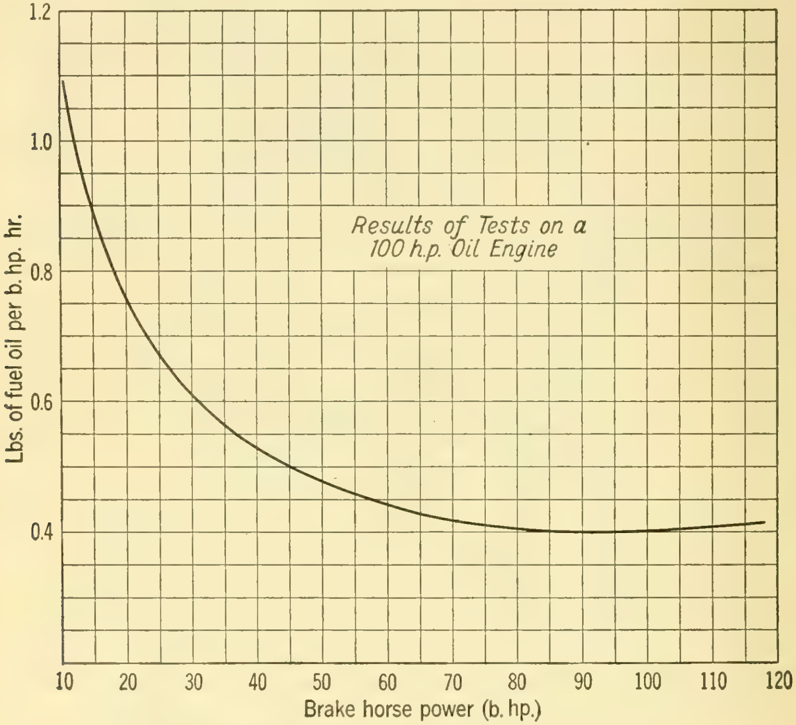


Fig. 398.—Worthington Oil Engine Economy Curve.

cal, the saving figured from a consideration of the fuel and operator costs must pay for the additional costs within a reasonable time.

Figures 396 and 397 give details of different designs of oil engine cylinders.

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